Optical investigation of propane-dimethyl ether (DME) fuel blends under compression-ignition engine conditions

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Abstract

Propane, DME, and propane-dimethyl ether (DME) blends were investigated in an optically accessible engine under compression ignition (CI) engine conditions. Schlieren imaging was used to provide information on the physical development of fuel jets under non-reacting conditions. OH* chemiluminescence imaging and in-cylinder pressure data were used to assess the combustion characteristics of propane, DME, and propane-DME blends varying from 10% to 30% DME by mass. First, near-top dead center (TDC) single-injection experiments were carried out to examine the impact of injection timing, intake temperature, and injected fuel energy on the ignition and combustion process. Ideal start of injection (SOI) timing was determined based on maximizing indicated efficiency and varied with DME concentration from -13.25 CAD for propane and -8.25 CAD for the 70% propane-30% DME blend. Below an intake temperature of 180 °C, combustion efficiency significantly decreased for propane and the 90% propane-10% DME blend. The lack of mixing-controlled compression ignition (MCCI) combustion using a single-injection strategy necessitated a dual-injection strategy. For the dual-injection experiments, pilot SOI timing was swept from -50 CAD to -20 CAD, and main injection timing was held constant at -5 CAD. Propane only exhibited MCCI behavior at the most delayed pilot SOI, while the 70% propane-30% DME blend exhibited it at all pilot SOI timings. Lastly, using a similar dual-injection strategy, a continuously operated glow plug was used to assist ignition at a reduced intake temperature and achieve reliable mixing-controlled combustion. Under low load and compressed gas temperatures representative of cold start conditions, the combustion of the fuel jets nearest the glow plug was reliable, but the ignition of the fuel jets far from the glow plug was improbable, and combustion efficiency was low. Increasing fueling led to better azimuthal propagation of the pilot combustion and increased momentum of the pilot combustion products, positioning them closer to the injector before the main injection. This facilitated the autoignition of more fuel jets compared to the lower load case and an appreciable increase in combustion efficiency.
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Chapter 1

Introduction

1.1 Motivation and background

The transportation sector significantly contributes to climate change, responsible for 28% of greenhouse gas (GHG) emissions. This sector heavily relies on petroleum-based fuels, which account for over 94% of its energy consumption \[1\]. Projections indicate a continued rise in energy usage for transportation, with estimates suggesting this trend will persist until 2030 \[2\]. In the United States, the reliance on internal combustion engines (ICEs) for power is striking, with over 250 million highway vehicles operating on this technology \[3\].

Given these statistics, it is evident that along with adopting zero-tailpipe vehicle technologies, reducing emissions from ICEs should be a primary focus in combating GHG emissions. Compression-ignition (CI) engines present a promising solution, offering higher thermal efficiency, faster torque response, and greater torque at lower speeds than spark-ignited (SI) engines. These characteristics make CI engines attractive for various applications, including medium- and heavy-duty on-road vehicles and off-road equipment that must meet demanding tractive and hydraulic combined loads.

CI engines generally utilize diesel fuel, leading to a combustion process often called conventional diesel combustion (CDC). While CDC in CI engines boasts superior efficiency
compared to SI combustion, along with better torque response and the ability to generate high torque at low speeds, there are notable drawbacks associated with CDC, one of which is the fuel itself.

Diesel fuel is predominantly derived from crude oil, a resource that demands substantial energy for extraction, transportation, refining, and subsequent delivery to filling stations for distribution. Importantly, each step contributes to the "well-to-tank" greenhouse gas (GHG) emissions before the fuel is ignited. The carbon intensity of a fuel considers not only these well-to-tank emissions but also the energy-specific CO$_2$ emissions linked to combustion. Calculating the CO$_2$ emissions from combustion becomes straightforward when the chemical formula and lower heating value (LHV) of the fuel are known

\[
gCO_2/MJ = \frac{C# \times MW_{CO_2}}{LHV \times MW_{fuel}}
\]  

(1.1)

Assuming an average chemical formula of $C_nH_{1.87n}$ and an LHV of 42.7 MJ/kg for diesel fuel [4], the energy release specific emission is 74.2 gCO$_2$/MJ. Propane and dimethyl ether (DME) have energy release specific emissions of 64.6 and 66.1 gCO$_2$/MJ, respectively. Propane and DME emit less CO$_2$ than diesel when burned and have lower well-to-tank emissions [5]. In fact, according to the California Air Resources Board (CARB), DME can be produced renewably from dairy biomethane with a carbon intensity of -278 gCO$_2$/MJ [6].

In practice, the efficiency of engines and powertrains significantly impacts CO$_2$ emissions. Modern propane-fueled engines operate on SI combustion and tend to have lower efficiencies than modern diesel engines, thereby limiting the CO$_2$ reduction potential. One approach to enhancing efficiency is leveraging the CDC process with propane fuel. However, propane is significantly less reactive than diesel fuel (its cetane number (CN) is undefined), making it a suboptimal choice for CI engines.

In contrast, DME boasts a CN of 55-60 [7] and burns cleanly, producing minimal particulate matter (PM), which makes it an excellent CI fuel. Dimethyl ether, like propane,
liquefies under moderate pressures, and the two fuels readily mix. Although propane-DME blends with high DME content exhibit superior performance in CI engines, DME’s corrosive nature towards common elastomers and significantly lower viscosity than diesel fuel raise concerns about potential fuel system damage. Moreover, the near-term availability of renewable DME is expected to be limited, necessitating the ability of propane-DME engines to operate reliably on blends with minimal or no DME content.

Given these challenges, it is crucial to develop a comprehensive understanding of the combustion processes of propane and propane-DME blends under CI-engine in-cylinder conditions. This work aims to provide insights for developing CI combustion strategies using propane and propane-DME blends as fuels. The analysis involves examining in-cylinder pressure data and employing imaging techniques in an optically accessible engine.

Specifically, schlieren imaging is utilized to study the physical evolution of fuel jets under non-reacting conditions by visualizing density gradients. Additionally, hydroxyl radical (OH*) chemiluminescence is employed to qualitatively identify high-temperature ignition regions. The study will explore the combustion process’s dependence on factors such as injection timing, intake temperature, and injected fuel energy. Furthermore, the potential use of a glow plug ignition assistant will be evaluated to facilitate the effective use of propane and propane-DME blends in CI engines.
Chapter 2

Literature review

This chapter reviews previous literature relevant to the results presented in this study. It will discuss past research on propane-DME blends in CI engines and under CI engine conditions. The next section will examine work done on the development of high-pressure fuel jets, beginning with a description of the spray regime under which CIDI fuel jets fall. Following that, there will be a review of the imaging techniques used in this study, emphasizing their application in engine research. Finally, the use of glow plugs in CI engines will be discussed, covering their use as intermittent cold start ignition assistants and their continuous operation to aid in igniting low-reactivity fuels.

2.1 Propane-DME in CI engines

Propane has an octane rating of 104-112 \[8\], making it a suitable SI engine fuel and a poor CI engine fuel. However, there has been research interest in using propane and propane-DME blends in CI engines due to the lower carbon intensity and clean-burning properties relative to diesel fuel. Kajitani et al. \[4\] studied the effects of propane blend percentage in DME on combustion stability, fuel consumption, and emissions in a typical CIDI engine with a compression ratio of 17.7. They concluded that the maximum mass percentage of propane blended into DME while still achieving combustion on their setup was 60\%, above which
there was no combustion, as shown in Figure 2.1. They also found that as the percentage of propane increased, injection timing had to be advanced to maintain stable combustion due to increasing ignition delay (ID). Lastly, they found that NO\textsubscript{x} emissions decreased with increasing propane percentage.

![Figure 2.1](image)

**Figure 2.1:** Figure from [4] showing the operating region of the DI diesel engine with propane-DME fuel blends.

Mochizuki [9] et al. conducted homogeneous charge compression ignition (HCCI) experiments with propane-DME blends in an optically-accessible single-cylinder engine. A formaldehyde absorbance technique determined that low-temperature oxidation increased with increasing propane blend percentage. Chapman and Boehman [10] conducted "mixed mode" combustion process experiments with DME and diesel where DME was fumigated into the intake air along with a near-TDC single, direct diesel injection. The authors showed reductions in NO\textsubscript{x} emissions and increases in energy conversion efficiency at DME energy fractions up to 44% but noted an increase in particulate emissions with DME fumigation due to the displacement of oxygen with DME leading to higher local equivalence ratios in the diesel spray flames. Furthermore, the advanced combustion phasing of the premixed charge led to high peak cylinder pressures that the authors posit as potentially harmful to light-duty diesel engines.
Mohammed et al. [11] conducted ignition delay time (IDT) measurements at MCCI-relevant engine conditions in a shock tube using neat propane, neat DME, and a 60% DME-40% propane blend by mole fraction. Figure 2.2 shows the IDTs as a function of temperature and fuel composition for the three mixtures. All experiments were carried out at an equivalence ratio ($\phi$) of 1. The results highlight the relatively long IDT of neat propane relative to DME and the DME-propane blend and the effect of higher pressures reducing the IDT for propane and the DME-propane blend.

Figure 2.2: Figure from [11] showing the IDTs of neat propane (mix 1), neat DME (mix 2), and the 60% DME-40% propane (mix 3) at 60 and 80 bar over a range of temperatures.

2.2 Fuel jet development

The CDC process relies heavily on the development of the high-pressure liquid fuel injected into the combustion chamber near TDC. In general, liquid jets are classified by the mechanism of spray breakup into four regimes: Rayleigh breakup, first wind-induced breakup,
second wind-induced breakup, and atomization \cite{12}. Figure 2.3 displays images of liquid jets in each of the four regimes.

Figure 2.3: Figure from \cite{12} showing (a) Rayleigh breakup, (b) first-wind induced breakup, (c) second-wind induced breakup, and (d) atomization.

These four regimes are classified by the non-dimensional Weber (\(W_e\)) and Ohnesorge (\(Oh\)) numbers defined as:

\[
W_e = \frac{\rho_a U^2 d}{\sigma} \quad \text{(2.1)}
\]

\[
Oh = \frac{\mu}{\sqrt{\rho_f d \sigma}} \quad \text{(2.2)}
\]
where ρa and ρf are the ambient gas and fluid densities, respectively, U is the injection velocity, d is the orifice diameter, σ is the surface tension, and μ is the liquid dynamic viscosity. Fuel jets in CIDI engines typically fall into the atomization regime depicted in Figure 2(d) where We > 40.3 and spray breakup starts at the nozzle exit [12,13]. The tiny, atomized liquid droplets promote fast vaporization of the liquid fuel, reducing the time between the start of injection (SOI) and ignition. The primary breakup mechanism for atomizing sprays is still uncertain [14], and capturing the initial atomization in experiments is complicated by small orifice diameters (100-200 µm), high injection velocities, and large optical depths [15]. Conversely, macroscopic features (e.g., jet penetration and dispersion) of high-pressure fuel jets are relatively easy to capture in experiments and have been studied since the 1920s [16,17].

Hiroyasu and Arai [18] measured breakup length, dispersion, penetration, and drop size distribution of high-pressure diesel jets. The breakup length was defined as the contiguous portion of the liquid column of the fuel jet. The authors noted two distinct breakup regimes with respect to injection velocity. At low injection velocities, breakup length increased with injection velocity, but as velocity increased, breakup length decreased as velocity increased, after which the breakup length remained relatively constant. The latter regime encapsulated DI diesel sprays with high injection velocities. Naber and Siebers [16] examined the effects of ambient gas density on the injection and vaporization of diesel fuel jets in a constant-volume combustion chamber. Non-vaporizing and vaporizing conditions were studied at various ambient densities, temperatures, injection pressures, and orifice diameters. The authors measured the penetration distance (S) and the full cone angle (i.e., dispersion angle) (θ) using a schlieren imaging system and processing routine. Figure 2.4 shows how the sprays were characterized using penetration distance and dispersion angle.

The dispersion angle, θ, is defined as:

\[ \frac{\theta}{2} = \tan^{-1} \left( \frac{A_{p,S/2}}{(S/2)^2} \right) \]  

(2.3)
where $A_{p,S/2}$ is the projected area of the upstream half of the spray. The local dispersion angle, $\theta_l(x)$, at different axial locations was found to reach the same quasi-steady value $\sim 0.5$ ms after the spray head reached each location, showing that the local dispersion angle is independent of axial location and that the entrainment rate along the jet is constant. For non-vaporizing sprays, the dependence of the tangent of the dispersion half-angle on the ratio of ambient gas and injected fuel densities was $(\rho_a/\rho_f)^{0.19}$. The dispersion of vaporizing sprays did not follow the same relationship, especially at lower ambient densities, as illustrated in Figure 2.5.

The authors noted that the reduction in dispersion angle caused by fuel vaporization is most likely due to the cooling and subsequent contraction of the entrained hot gases as fuel is evaporated. Penetration results followed a similar trend to the dispersion results with respect to vaporization. The penetration results from Naber and Siebers [16] did not agree with previously developed penetration correlations from Hiroyasu and Arai [18]. Naber and Siebers developed characteristic length and time scales and a non-dimensional penetration correlation following an analysis by Wakuri [19]. The characteristic length ($x^+$) and time
Figure 2.5: Figure from [16] showing data points for all vaporizing conditions and fits to the non-vaporizing data.

\((t^+)\) scales are:

\[ x^+ = \frac{d_f \cdot \sqrt{\rho}}{a \cdot \tan(\theta/2)} \]  

\[ t^+ = \frac{d_f \cdot \sqrt{\rho}}{a \cdot \tan(\theta/2) \cdot U_f} \]  

where \(\tilde{\rho}\) is the ratio of the fuel density to the ambient density, \(U_f\) is the fuel velocity at the orifice exit, and \(a\) is a constant that relates an actual spray to the model spray. The correlation for the penetration time versus distance:

\[ \bar{t} = \frac{\tilde{S}}{\tilde{t}} + \frac{\tilde{S}}{2} \cdot \sqrt{1 + 16 \cdot \tilde{S}^2} + \frac{1}{16} \ln \left( 4 \cdot \tilde{S} + \sqrt{1 + 16 \cdot \tilde{S}^2} \right) \]  

and the approximate inverse for penetration distance versus time is:

\[ \tilde{S} = \left( \frac{1}{\tilde{t}} \right)^n + \left( \frac{1}{\tilde{t}^{1/2}} \right)^n \]  

where \(n = 2.2, \tilde{t} = t/t^+, \) and \(\tilde{S} = S/S^+\).
The characteristic length and time scales given by Equations 2.4 and 2.5 collapsed all non-vaporizing data to within 10% of the correlation given by Equation 2.6, and all vaporizing data to within 18%.

2.3 Optical engine measurement techniques

This section provides a literature review of the two imaging techniques used in this work on an optically-accessible engine. First, schlieren imaging and its applications in engine research in the literature will be discussed, followed by a review of OH* chemiluminescence imaging in engines.

2.3.1 Schlieren imaging

Schlieren imaging is a technique used to visualize fluid flow that was invented by August Toepler over a five-year period between 1859 and 1864 [20]. The technique relies on changes in the index of refraction due to density gradients in a flow to create images of transparent and optical media [21]. Figure 2.6 from [22] illustrates a typical schlieren imaging arrangement using an aperture as the cutoff.

There have been many efforts to characterize engine flows using schlieren imaging. Namazian et al. [23] used a "Z type" schlieren imaging configuration to visualize the flow and density fields in an SI engine with a square piston and parallel quartz windows on opposite sides of the rectangular prism that formed the "cylinder" which provided optical access perpendicular to the "cylinder" axis. The imaging setup resolved the vortices around the edges of the intake valve as well as crevice flows during the intake and expansion strokes. The spark plug was located on the side of the rectangular prism wall, and the authors measured apparent flame velocity by tracking the flame front in successive images as it traveled across the combustion chamber.

Schipperijn et al. [24] used schlieren imaging, among other techniques, such as planar
laser-induced fluorescence (PLIF) and Mie scattering to observe liquid fuel sprays, flame propagation, and qualitative aspects of fuel-air mixing during the intake stroke. Schlieren images resolved provided signal intensity variations in both the liquid and vapor regions of fuel spray, while Mie scattering showed only the liquid portion of the spray. These images confirmed that the vapor phase of the spray was downstream of the liquid droplets, as expected.

Schlieren-based line-of-sight (LOS)-averaged temperature measurements were conducted by Aleiferis et al. [25] in an optically-accessible SI engine in the reaction zone of an iso-octane and air mixture. Using the Gladstone-Dale relation and the ideal gas law, the authors formulated an equation for LOS-averaged temperature given by:

$$\frac{\partial \langle T \rangle}{\partial y} \frac{1}{\langle T \rangle^2} = \pm \frac{I_d - I}{I} \frac{\alpha_k n_0}{f L(x)} \frac{\rho_0}{n_0 - 1} \frac{R}{p}$$  \hspace{1cm} (2.8)$$

where $\langle T \rangle$ is the LOS-averaged temperature along $L(x)$, $n_0$ and $\rho_0$ are the reference refractive index and density, $p$ is the absolute pressure, $\alpha_k$ is the dimension of the undisturbed beam above the knife edge, $f$ is the focal length of the focusing lens, and $I$ and $I_d$ is the illumination...
of the undisturbed and disturbed field, respectively. Equation 2.8 was integrated numerically to solve for the LOS-averaged temperature, \( \langle T \rangle \). Figure 2.7 shows a schlieren image and LOS temperature distribution at 20 CAD after ignition.

Figure 2.7: Figure from [25] showing a schlieren image and the corresponding LOS temperature distribution at 20 CAD after ignition for an SI engine operating with iso-octane at an air-to-fuel ratio of 15.

Neal and Rothamer [26] utilized shadowgraphy to measure the penetration and dispersion of transient diesel fuel jets in an optically-accessible engine resembling a light-duty CIDI engine. Shadowgraphy is similar to schlieren imaging, but while schlieren imaging detects changes in the first derivative of density, shadowgraphy detects changes in the second derivative of density. Shadowgraphy systems also do not use a cutoff and are generally less sensitive to changes in density compared to schlieren systems. An example of jet images using shadowgraphy from [26] is shown in Figure 2.8.

Penetration results from the shadowgraph images contained three unique regions in time with different penetration scaling with time. The first region was the initial transient region close to SOI where penetration was found to scale with \( t^{1.4} \). In between the transient region and the far-field region, a transition region was identified where, depending on injection pressure, penetration was found to scale with \( t^{0.8 - 1.1} \), with the exponent increasing with injection pressure. In the far-field region, the penetration scaled with \( t^{0.5} \), which agrees with previous quasi-steady state jet studies [16] [18].
Figure 2.8: Shadowgraphy images of diesel fuel injected into an ambient density of 15 kg/m$^3$ at an injection pressure of 30 MPa [26].

2.3.2 OH* chemiluminescence

OH* chemiluminescence is the spontaneous light emission from the excited hydroxyl radical (OH*) during a typical hydrocarbon combustion reaction. The excited hydroxyl radical is formed primarily through the reaction \( \text{CH} + \text{O}_2 \rightarrow \text{CO} + \text{OH}^* \) [27] and emits at a characteristic wavelength of approximately 310 nm. There are many other excited radicals that form during combustion and chemiluminesce including \( \text{CH}^* \), \( \text{C}_2^* \), and \( \text{CH}_2\text{O}^* \), but OH* has been found to better represent flame fronts and structures than other emitting radicals [28]. OH* chemiluminescence imaging and spectroscopy are commonly employed in optically accessible engines and combustion chambers to qualitatively identify high-temperature reaction regions [29] and to measure quantities such as lift-off length [26, 27] and local equivalence ratio [30, 31].

White [30] used OH* chemiluminescence imaging in an SI hydrogen-fueled optical engine to measure flame propagation characteristics and to develop a functional relationship between OH* chemiluminescence intensity, cylinder pressure, and equivalence ratio. Figure 2.9 illustrates the relationship between OH* chemiluminescence intensity, global equivalence ratio, and cylinder pressure.

The surface in Figure 2.9 is of the form
Figure 2.9: Mean OH* chemiluminescence intensity for pre-IVC injection data sets plotted against pressure and equivalence ratio. The surface is the functional relationship from a non-linear least squares fit to the data.

\[ I = AP^B \exp(CP^D\phi) \]  

(2.9)

where \( A, B, C, \) and \( D \) are fitting constants. The author rationalized using pressure as an independent variable arguing that since cylinder temperature is strongly correlated to cylinder pressure and flame temperature is a function of reactant temperature, stoichiometry, and pressure, the dependence of OH* chemiluminescence on flame temperature is encompassed by the dependence of OH* chemiluminescence on pressure and equivalence ratio.

Higgins and Siebers [27] conducted lift-off length measurements of reacting diesel sprays using OH*, CH*, and C\(_2\)* chemiluminescence imaging in a constant volume combustion chamber over a wide range of ambient gas densities, temperatures, and oxygen concentrations. The authors concluded that OH* chemiluminescence provided the best marker of lift-of-length due to its strong chemiluminescence at 310 nm far from broadband soot incandescence. They also found that as the combusting spray releases heat and chamber temperatures rise, the lift-off length progressively shortens as depicted in Figure 2.10 where
the average pressures during the camera gate period for images (a), (b), and (c) were 2.39, 2.25, and 2.52 MPa, respectively. Using an isentropic compression assumption, the gas temperatures at these pressures were 1040, 1024, and 1053 K, respectively.

Figure 2.10: Three images of time-averaged OH* chemiluminescence emission. The camera intensifier gate duration and start time after SOI (delay) are reported in the top left corner of each image. The graph on the bottom displays the axial profiles of the relative OH* chemiluminescence intensity for the images [27].
2.4 Glow-plug ignition

Energy-assisted ignition in the context of engines refers to the local deposition of energy into the combustion chamber to assist the ignition process. Glow plugs have been used in CI engines to assist with ignition and reduce emissions under cold-start conditions [32–34], in continuous operation to assist with the ignition of low reactivity fuels [29, 35, 36] in MCCI engines, and to control combustion in homogenous charge compression ignition (HCCI) engines [37–39].

Lindl and Schmitz [33] compared the effects of glow plugs and intake air heating using electric heaters and intake burners on cold start hydrocarbon (HC) emissions and run-up time. They concluded that for small- and medium-sized diesel engines (200-1000 cm$^3$ cylinder displacement), the glow plug demonstrated reliable cold start comparable to SI engines and significantly reduced HC emissions above a glow plug temperature of 1100 K. Figure 2.11 illustrates the effect of glow plug temperature on HC emissions and run-up time to a stable engine speed during cold start. For engines with piston displacements above 1500 cm$^3$ at ambient temperatures below -30 °C, the authors concluded that the only way to achieve a reliable cold start was to use an intake manifold burner.

![Figure 2.11: HC emissions and run-up time versus glow plug temperature at an intake air temperature of -20 °C](image)

Figure 2.11: HC emissions and run-up time versus glow plug temperature at an intake air temperature of -20 °C [33].
Kobayashi et al. [32] used sheathed-type electric glow plugs in each cylinder of an inline 4-cylinder truck engine and compared the cold start characteristics to the stock engine without glow plugs. Results at an ambient temperature of -25 °C indicated that the glow plug system reduced the starting period from 80 seconds to 55 seconds at an injection timing of 14 degrees before TDC. Mueller and Musculus [36] conducted optical engine experiments examining glow plug-assisted ignition of single injections of methanol. The glow plug was shrouded to help maintain temperature and controlled through a unique relationship between temperature and resistance. Through heat release rate and direct luminosity image analysis, three types of heat release rate profiles were identified for individual cycles, as shown in Figure 2.12.

![Figure 2.12](image)

Figure 2.12: Three types of apparent heat release rate (AHRR) traces observed in individual cycles during methanol combustion. (a.) The jet-pair numbering system used to label AHRR features. (b.) Most probable pair-wise jet ignition, (c.) Mixed pair-wise and single-jet ignition second most probable, and (d.) All single-jet ignition is least probable [36].

Amezcua [40] utilized OH* chemiluminescence imaging and demonstrated the influence
that custom piston bowl design has on glow plug-assisted compression ignition of alcohol-to-jet fuel with a derived cetane number (DCN) of 17 in an optically accessible engine. Figure 2.13 shows a schematic of the custom piston bowl (left) design relative to the baseline bowl (right).

![Figure 2.13: Custom piston bowl with "waves" (left) and baseline piston bowl (right). The red dashed circle at the center of the bowl represents the location of the fuel injector, and the red dashed oval below the injector represents the location of the glow plug [40].](image)

The same dual-injection strategy was used for both bowl designs. OH* chemiluminescence of the pilot-only combustion for both bowl geometries is shown in Figure 2.14. The custom piston bowl redirected the jet injected at the glow plug toward the center of the bowl and positioned the combusted gasses around the injector before the main injection, which allowed the main injection to ignite rapidly and transition to MCCI.
Figure 2.14: OH* chemiluminescence images of the glow plug-assisted combustion process using the (a) baseline piston bowl and (b) custom piston bowl for a 7 mg pilot injection with an SOI of -26.75 CAD. The red dashed circle and oval represent the injector and glow plug, respectively [40].
2.5 Literature review summary

The literature has not extensively covered the use of propane-DME blends in MCCI engines because of propane’s low reactivity. The limited work that has been done has not focused on the use of pure propane or blends with less than 40% DME by mass. However, there have been recent advancements in developing chemical kinetic mechanisms for propane, DME, and propane-DME blends under CI engine-relevant conditions in a shock tube [11].

The physical development of fuel jets has been extensively explored in the literature. Studies on spray breakup have shown that fuel jets in CIDI engines fall in the atomization regime with fuel beginning to atomize immediately at the nozzle exit. The exact mechanism for breakup in atomizing sprays is still uncertain, but the macroscopic qualities of quasi-steady sprays downstream of the nozzle exit have been well studied. Importantly, the characteristics of jet penetration and dispersion, which determine air entrainment and local equivalence ratio, have been found to depend on the ratio of ambient density to fuel density.

Schlieren and OH* chemiluminescence imaging have been well documented in the literature as tools to capture in-cylinder engine phenomena. Schlieren imaging has mainly been used to analyze in-cylinder flows such as intake charge motion and fuel injection, as well as temperature measurements. OH* chemiluminescence imaging has been used to qualitatively identify regions of high-temperature ignition, determine the start of combustion, measure lift-off length, and provide semi-quantitative equivalence ratio measurements.

Glow plugs have long been used as cold-start ignition assistants in CI engines to reduce run-up time and HC emissions. More recently, there has been interest in using continuously operated glow plugs to assist the ignition of low-reactivity fuels like methanol and sustainable aviation fuels. Results indicate that the glow plug-assisted combustion process is primarily driven by surface temperature and piston bowl geometry.

While propane has been studied as an SI engine fuel, there is a lack of research on utilizing propane and blends of propane-DME with low DME content in CI engines. While the imaging techniques used in this work are not novel, they have not been used to study
the jet development and ignition behavior of the propane-DME blends in an optical engine and can provide valuable insights into the combustion process.
Chapter 3

Experimental setup, operating conditions, and methods

3.1 Optical engine laboratory

The ERC optically-accessible CIDI engine laboratory, or optical engine lab for short, is built around a modified single-cylinder General Motors Research Laboratory engine that accommodates a Bowditch \cite{41} style piston, which allows for optical access into the combustion chamber. The engine configuration is highly flexible allowing for rapid changes to be made to the combustion chamber, instrumentation, and the imaging techniques and optical diagnostics being employed. Over the last several years, a significant amount of research has been conducted in the optical test cell on a wide range of projects and topics. Each project necessitates a unique lab configuration so I will only describe the details of the test cell that were pertinent to the present work. For other descriptions of the optically accessible engine lab, refer to the works of Amezca \cite{40}, Groendyk \cite{42}, and Cherry \cite{43}.
### 3.1.1 Engine specifications

The geometry of the optical engine is based on a typical light-duty automotive diesel engine. Table 3.1 details the important characteristics of the engine.

<table>
<thead>
<tr>
<th><strong>Table 3.1: Engine parameters.</strong></th>
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<tbody>
<tr>
<td>Compression ratio</td>
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<tr>
<td>Displacement [L]</td>
</tr>
<tr>
<td>Stroke [mm]</td>
</tr>
<tr>
<td>Bore [mm]</td>
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<tr>
<td>Con-rod length [mm]</td>
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<tr>
<td>Intake Valve Closing [°aTDC]</td>
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<td>Exhaust Valve Opening [°aTDC]</td>
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<td>Intake Valve Opening [°aTDC]</td>
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<td>Exhaust Valve Closing [°aTDC]</td>
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</tbody>
</table>

The compression ratio reported in 3.1 is not geometric, rather it was calculated from motored pressure data by minimizing the error of linear least squares fits to the decimal logarithm of the pressure as a function of the decimal logarithm of the volume (log(P) vs. log(V)) of the compression and expansion strokes. Figure 3.1 is an example of a typical motored log(P) vs. log(V) plot with linear least squares fits to the compression and expansion strokes and the corresponding fit residuals for a motored run with a cycle averaged peak pressure of 30.7 bar.

Figure 3.2 shows cross-sectional views of the engine with cutting planes parallel to the cylinder axis and at the top surface of the piston window. Important components and features of the engine and combustion chamber are labeled. The piston bowl geometry is a right-cylinder as opposed to the conventional re-entrant bowl design used on many diesel engines. The flat surfaces of the piston window allow light rays incident on the top surface of the window to exit the bottom surface of the window parallel to the ray incident on the top surface and vice versa, meaning that images of the combustion chamber are undistorted. The piston window has an outer diameter of 59.94 mm and a height of 28.96 mm. The top and bottom sides of the window are optically polished with a scratch-dig specification of 20-10
Figure 3.1: (a) log(P) vs. log(V) for a cycle-averaged (200 cycles) motored pressure trace with linear least squares fits to the compression and expansion strokes. (b) Fit residuals vs. log(V) for compression and expansion strokes.

Figure 3.2: (a) Cross-sectional rendering of the optical engine with the cutting plane parallel and coincident with the cylinder axis and (b) a cross-sectional view of the combustion chamber with the schlieren imaging and OH\(^*\) chemiluminescence field-of-views outlined.
and a wavelength accuracy of $1/2 \lambda$. The cylindrical walls of the window are ground, and an epoxy adhesive (3M™ DP460NS) is used to secure the piston window into the piston cap, which has an inner diameter of 60.96 mm. The squish ring, which forms the piston bowl, is fastened to the piston cap with four 2-56 thread socket head cap screws. A 1/32" thick annular gasket (KLINGERSIL® C-4401) is adhered to the portion of the squish ring that overlaps with the piston window using room-temperature-vulcanization (RTV) silicone to prevent metal-glass contact. The inner diameter of the squish ring is 53.33 mm and forms the OH* chemiluminescence field-of-view (FOV).

The optical engine is run dry to prevent oil droplets from landing on the piston window and obstructing the FOV. Of course, below the piston extension, oil circulates through the crankcase and lubricates the base engine crankshaft bearings and piston-cylinder interface, but no oil from the crankcase lubricates the optical piston. A vacuum pump evacuates the crankcase to prevent oil from flowing above the piston rings and ending up on the piston window. The combustion chamber is sealed using a custom Teflon™ sealing ring (Saint Gobain, Omniseal 223-234-A05-04). The inside of the sealing ring is filled with RTV silicone to reduce crevice volume. A Vespel® rider ring is used to ensure the piston remains concentric with the cylinder and to prevent piston-cylinder contact during operation. A felt ring is impregnated with a small amount of lightweight oil to provide some lubrication. Figure 3.3 shows a close-up view of the piston-cylinder interface and the rings described above.

Engine run-time is recorded and the sealing ring is replaced every 1200 minutes due to wear. The felt ring is frequently checked and oil is added when it becomes dry. Each time the piston cap is removed, new screws are used to secure the piston cap as a safety measure.

### 3.1.2 Glow plug specifications and spray targeting

The glow plug (GP) study was performed with a ceramic GP (Beru CGP003). The GP was operated using a remotely controlled power supply (B&K Precision 9115), maintaining
Figure 3.3: Close-up cross-section of piston cylinder interface and the rings used to seal, lubricate, and guide the piston extension in the cylinder.
a constant voltage set to achieve the required power and a current limit of 7 Amps. The recommended input voltage for this GP is 7 V, typically yielding a power output of 30 W. However, for this specific application, it was operated at a higher power level of 75 W. Amezcua et al. 29 measured the surface temperature of this GP on the optical engine while motoring at a range of power levels and based on the power-law fit to their data, the surface temperature of the GP at 75 W is 1450 K.

The ignition and combustion process with GP-assisted ignition is highly sensitive to the interaction between the GP and the fuel jet, which is directed toward or near the GP 29 36. In this study, the fuel injector was positioned such that one of its seven holes was aligned directly with the GP. This alignment was achieved by imaging the fuel jet using a schlieren system with the GP off, the engine stationary and the exhaust valve open. Figure 3.4 illustrates the alignment of the fuel jet in line with the GP.

3.1.3 Control and data acquisition

Due to the lack of lubrication, the optical engine is only run for approximately 5-7 minutes at a time, followed by a 20-minute cool-down period. Each run is timed with a stopwatch. For the first 5 minutes, the engine is motored (turned by the dynamometer while no fuel is injected) to establish quasi steady-state intake and in-cylinder thermodynamic conditions 44. After the 5-minute motoring period, the engine is fired (fuel is injected) at a skip-fire ratio (SFR) of 9:1 (9 motored cycles followed by 1 fired cycle) for 200 total cycles and 20 fired cycles. Skip-fire operation, combined with the limited number of fired cycles, reduces thermal loading from combustion gasses on the piston window and maintains acceptable piston surface temperatures.

The intake flow rate is precisely controlled using a choked flow orifice rack that was calibrated using a Coriolis flowmeter. A 2.8-kW PID-controlled inline process air heater and downstream tape heaters are used to precisely heat intake air to temperatures up to 200°C. Intake pressure is measured on a crank-angle resolved basis with a high-speed, piezoresistive
Figure 3.4: Fuel jet positioning relative to the GP: (a) Schlieren image of a propane fuel jet viewed through piston window. The large red arc on the right corresponds to the piston bowl and the smaller red oval is the hole through which the glow plug protrudes. (b) CAD cross-sectional side view of the combustion chamber at TDC showing a fuel jet with a 22° dispersion angle interacting with the GP.
pressure transducer (Kulite XTEL-190-50A). Intake temperature is monitored using a type-K thermocouple. A piezo-electric pressure transducer (Kistler 6053BB) coupled to a dual-mode amplifier (Kistler 5010) is used to measure in-cylinder pressure on a 0.25 crank-angle-degree (CAD) basis.

3.1.4 Data processing

Pressure

Engine data were saved in a custom binary file format (.xfml) detailed in [42]. The .xfml file structure is designed to contain recorded data alongside instrumentation uncertainty information, which enables straightforward uncertainty propagation. This is achieved using an object-oriented post-processing routine developed by Joshua Herzog. Cylinder pressure data is sampled at encoder resolution (0.25 CAD), and 200 cycles (20 fired, 180 motored) are saved for each run. To remove signal noise and high-frequency ringing from pressure data, raw pressure was filtered using a two-pass, 2nd order low-pass Butterworth filter with a 3 kHz cutoff frequency. The heat release rate (HRR) was calculated from the filtered pressure traces according to

\[
\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} + \frac{dQ}{d\theta_{HT}}
\]

(3.1)

where \(\frac{dQ}{d\theta_{HT}}\) is the rate of heat transfer from the gas to the in-cylinder surfaces estimated using the Woschni correlation [45], and \(\gamma\) is the heat capacity ratio evaluated at the estimated isentropic compressed gas temperature using an exponential fit to data from [46]. Derivatives were solved numerically using the central difference method. The HRR calculations are heavily impacted by the method of pressure filtering used. However, when consistent filtering techniques are applied, the HRR values should be reliably affected.

Since emissions measurements are not taken on the optical engine, the HRR cannot be scaled based on combustion efficiency. Therefore the HRR is not necessarily numerically
accurate, but it still is a very useful metric to determine combustion modes and to compare results from different experiments.

The start-of-combustion from pressure (SOC) was determined using the method detailed in [47]. This method subtracted the mean raw motored pressure from the raw fired pressure data and integrated the resulting difference. The location of the minimum integrated pressure difference after SOI was identified as the SOC. This metric demonstrates robustness when ignition delays are sufficiently long, leading to significant pressure depression from evaporative cooling. However, in cases with very short ignition delays (e.g., DME), the metric may inaccurately predict the SOC as occurring too early. This discrepancy arises because the pressure between SOI and SOC can be greater than the mean motored pressure due to cycle-to-cycle variability, leading to premature estimations of SOC. So, SOC values were verified, and any outliers were investigated and, if necessary, removed.

**Schlieren images**

Schlieren images were processed to calculate the penetration and dispersion of the fuel jet aligned with the polished surface and mirror mount. Figure 3.5 illustrates the processing method used to binarize the schlieren images. After scaling and binarization, penetration was calculated as the maximum location where the binarized jet was at least 4 pixels wide, and the dispersion angle was calculated using the method of Naber and Siebers [16], which is detailed in Section 2.2.

Certain portions of the fuel jet beneath both the injector and the dark zone created by the interface between the polished area and the precision mirror were not captured. To address this, a reconstruction method detailed in [44], was used to extrapolate and complete the missing regions of the jet as illustrated in Figure 3.6.
<table>
<thead>
<tr>
<th>Step 1: Frame differencing</th>
<th>Step 2: Apply 3 passes of 3 x 3 median filter</th>
<th>Step 3: Apply image threshold (Otsu’s method)</th>
<th>Step 4: Add new region to jet region image</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frame i</td>
<td>Frame i+1</td>
<td>Frame i – Frame i+1</td>
<td></td>
</tr>
</tbody>
</table>

Figure 3.5: Schlieren processing routine.

Figure 3.6: Example of jet image binarization and filling in dark zones.
**OH* chemiluminescence images**

OH* chemiluminescence images were processed to determine SOC (SOC$_{\text{OH}}$) utilizing an average intensity threshold method, as illustrated in Figure 3.7. The frame at which the average image intensity crossed the $\mu+5\sigma$ threshold was selected as the frame for determining SOC$_{\text{OH}}$. The OH* chemiluminescence camera trigger was synchronized with the crankshaft encoder, ensuring a straightforward relationship between frame number and CAD. This synchronization facilitated a correlation between the in-cylinder pressure measurements and the OH* chemiluminescence imaging.

![Figure 3.7: Example of SOC detection using OH* chemiluminescence images. The blue line denotes the average normalized intensity of a frame within a captured set of images. The orange dashed line, represents the average intensity threshold defined as the mean value of the average intensity of 50 frames before SOI (shaded in gray) plus five standard deviations about the mean.](image-url)

Frame differencing and binarization techniques were employed to offer qualitative insights into the ignition propagation within the piston bowl region, both for individual fired cycles.
and for cycle-averaged images. Frame differencing was a crucial step in enhancing the binarization process. It effectively eliminated the background, allowing for the isolation of the newly formed OH* from the rest of the image. These methods allowed for the visualization and analysis of the ignition process under different conditions. Otsu’s method [48] was used to binarize the frame-differenced images, and small areas of less than 8 pixels were removed. A schematic of the frame differencing and binarization process is shown below in Figure 3.8.

![Figure 3.8: Schematic representation of frame differencing and binarization approach used for OH* chemiluminescence images. Images acquired at 0.5 CAD, from the fuel energy sweep using the 70-30 blend.]

Cycle-averaging of the images was employed to effectively remove cycle-to-cycle variability, yielding a more reliable and comprehensive overview of the ignition process by providing averaged information across multiple cycles. Figure 3.9 below displays an example of a single-cycle image versus a cycle-averaged image, demonstrating the effect of this method. These images are from the fuel energy sweep utilizing a 70-30 blend at the maximum injected fuel energy. In these visualizations, the x and y-axes represent pixel locations.

The glow plug (GP) study utilized OH* chemiluminescence images to generate ignition probability maps. This was achieved by binarizing each image and then aggregating them, dividing by the total number of cycles. This approach effectively consolidates OH* chemiluminescence data from multiple cycles into a single stack of images. An example of the outcomes from this method can be seen in Figure 3.10 depicting a GP-assisted pilot-main
3.1.5 High-pressure fuel system

Due to low quantities of injected fuel and ample pressurized air in the optical engine lab, the optical engine fuel pump is a pneumatic-driven liquid pump (Haskel DSTV-150) that can hold fuel pressure without consuming power. The pump comes stock with fluoro-rubber (Viton™/FKM) seals in the wetted section, which has been shown to be incompatible...
with DME [49]. While Wu et al. [49] showed that ethylene propylene diene terpolymer (EPDM) rubber was compatible with DME, only ethylene propylene rubber (EPR) seals were available as an alternative to FKM. EPR is functionally similar to EPDM, so the pump was ordered with the EPR seal kit (modification -51331). An air amplifier system (High-Pressure Technologies AS-SPLV2-10GH) is used to boost compressed building air from 90-100 psig to 180 psig, which feeds the pump. The output fuel pressure is directly proportional to the air supply pressure, which is PID controlled by a digital pressure regulator (Proportion Air PSR4PN). Pneumatic-driven pumps utilize a differential piston, i.e., a large, air-driven piston acting directly on a smaller hydraulic piston. The ratio of the piston areas determines the hydraulic output pressure for a given air drive pressure. The pump used in the optical engine laboratory has a piston area ratio of 173:1, which boosts the output hydraulic pressure by a factor of 173 relative to the air drive pressure.

Since propane and DME are liquefied gasses, bleeding the fuel system safely was an important consideration when designing the fuel system. Fortunately, the building exhaust system was easily accessible and excess fuel was simply routed into the building exhaust as they both have low inhalation toxicities [50][51] and do not act as GHGs. Figure 3.11 shows a simple schematic of the fuel system.

The optical engine propane-DME fuel system utilizes a "dead head" design where low-pressure fuel does not recirculate back to the tank from the injector or common rail, as displayed in Figure 3.11. A dead head system significantly reduces the complexity of a liquefied gas fuel system because there is no need for back-pressure regulated return lines to the tank that would be required to keep the fuel liquefied. Most CIDI injectors use a solenoid-hydraulic system to inject fuel, which necessitates a low-pressure return line, but direct-acting injectors, such as the Delphi DFI3 used in this work, use a piezoelectric crystal stack to directly control injector needle motion and do not require a low-pressure return line.

The optical engine used in the experiments operates on a skip-fired principle, meaning that small amounts of fuel flow intermittently through the fuel system. This intermittent
Figure 3.11: Schematic representation of the propane-DME fuel system. Red lines indicate high pressure and green lines indicate relatively low pressure.

Flow pattern poses challenges for accurately measuring the fuel flow rate using a conventional flow meter. To overcome this challenge, the injected fuel mass of propane, DME, and their blends was measured by directing the fuel into a 440 mL spherical volume. This volume was instrumented with a high-precision pressure transducer (Omega MMA100V10B3A6T4A6) and a type-K thermocouple, as depicted in Figure 3.12. The determination of injection mass involved measuring the chamber’s mass before and after a 40 to 60-second injection period, conducted at 10 injections per second.

Figure 3.13 shows simple schematics of a conventional servo-solenoid injector (Delphi DFI1) and the Delphi DFI3 direct-acting injector used in this work. Directly controlling the needle in direct-acting injectors allows for fast response times after injection command and high needle velocities, leading to injection rate shapes that approach a square wave [52]. Information on driving the Delphi DFI3 in the optical engine laboratory in the ERC is provided in Appendix A.

The specific injector used in the present work was manufactured by Delphi under license and not produced for the aftermarket, so limited information about the injector nozzle ge-
Figure 3.12: Instrumented vessel used to measure injection mass.

Figure 3.13: Schematic from Dober et al. [52] of solenoid and direct-acting Delphi fuel injectors.
ometry, e.g., orifice diameter, included spray angle, and nozzle hole shape, was available. Since injector nozzle geometry has an important role in spray formation, the orifice diameter and included spray angle were measured. A digital imaging microscope was used to measure the orifice diameter. As shown in Figure 3.14, the magnified image of the orifice was automatically scaled to physical dimensions based on the magnification, and the diameter of the circle drawn over the orifice was taken as the orifice diameter.

![Figure 3.14: Magnified (10x) image of Delphi DFI3 injector orifice used to measure the orifice diameter.](image)

Imaging was also used to measure the included spray angle of the injector. A 0.0052" (132 µm) pin gauge was carefully fit into an injector orifice and imaged when deflected toward the top of the injector and when deflected toward the bottom of the injector. Images were taken at each deflection condition and processed using binarization and contour detection to measure the angle of the pin gauge relative to the injector. Figure 3.15 shows how the images were processed using example images. The injector was oriented vertically such that the slope of the pin gauge relative to the injector could be calculated directly using a linear least squares fit to the pin gauge contour. The included spray angle was calculated...
Figure 3.15: Graphic representation of the processing routine used to calculate included spray angle.

as $180^\circ - 2\theta$, where $\theta$ is the angle of the pin gauge from horizontal as shown in Figure 3.15.

The same procedure was applied to all seven of the injector orifices, and the average angle was taken as the included spray angle. Table 3.2 contains important characteristics of the injector used in this work.

Table 3.2: Injector parameters.

<table>
<thead>
<tr>
<th>Manufacturer/Model</th>
<th>Delphi/DFI3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of holes</td>
<td>7</td>
</tr>
<tr>
<td>Orifice diameter $[\mu m]$</td>
<td>139</td>
</tr>
<tr>
<td>Included spray angle [deg]</td>
<td>154</td>
</tr>
</tbody>
</table>

### 3.1.6 Cameras and optics

The configuration of optics and cameras in the optical engine laboratory changes and evolves depending on the requirements of the user. Recently, much of the work in the optical engine has been focused on characterizing and analyzing the combustion process of low-cetane-number fuels under CI engine conditions by utilizing simultaneous schlieren and OH* chemiluminescence imaging. The cameras, optics, and hardware used to perform these optical
measurements in the present work will be discussed in this section.

A schematic detailing the imaging setup used in this work is shown in 3.16. The schlieren imaging setup consists of a high-powered LED (Luminus CBT-120) that emits at a central wavelength of 462 nm with a full width at half maximum (FWHM) of 25 nm. The LED was powered by a DC power supply (B&K Precision 9115) operated in constant current mode at a current of 13 A. The LED emission was collimated using a Nikon NIKKOR 50-mm f/1.2 lens and directed towards the engine using a 30% reflectivity, 70% transmissivity broadband plate beamsplitter (Edmund Optics #48-900). The light then reflected off the 45° UV-enhanced aluminum mirror (Edmund Optics #32-135), passed through the piston window (Esco Optics ZP121AS1UV) into the cylinder and reflected off the precision-ground stainless steel mirror insert in the engine head back out of the cylinder. After exiting the engine, the light passed through a 500 mm focal length biconvex field lens (Thorlabs LA4246), and then a 460 nm center wavelength bandpass filter with a 50 nm FWHM (Omega Optical 460BP50) was used to isolate the LED wavelength and reduce the collection of other sources of combustion luminosity. A razor blade was mounted on a translation stage in front of the camera lens to cut off approximately 25% of the incoming light to provide ample sensitivity to in-cylinder density gradients. The 500-mm focal length field lens was positioned within 500 mm (approximately 475 mm) in path length from the engine head, creating a magnified virtual image of the engine head, which was imaged with a Nikon Micro-NIKKOR 200-mm f/4 lens onto the detector of a high-speed camera (Phantom v2640). Details about the camera, settings, and images are provided in 3.3.

The OH* chemiluminescence imaging system consisted of a 347-nm longpass beamsplitter used to reflect OH* chemiluminescence through a bandpass filter with a center wavelength of 310 nm and a FWHM of 10 nm (Asahi Spectra XPBA 310) to isolate the OH(A → X) emission which has a characteristic wavelength near 310 nm. The filtered light then passed through a UV lens (CERCO 100 mm f/2.8) onto a lens-coupled intensifier (Night Vision Systems, model 2071). The intensified signal was imaged with a Phantom v1840 high-speed
camera. The intensifier was synchronized to the camera trigger using a pulse-delay generator (Berkely Nucleonics Model 565).

Figure 3.16: Schematic of optical setup used to capture schlieren and OH* chemiluminescence images on the optical engine.

Table 3.3: Important camera and imaging parameters.

<table>
<thead>
<tr>
<th></th>
<th>Schlieren</th>
<th>OH* chemiluminescence</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Camera</strong></td>
<td>Phantom v2640</td>
<td>Phantom v1840</td>
</tr>
<tr>
<td><strong>Image plane pixel size [µm]</strong></td>
<td>27.5</td>
<td>27.5</td>
</tr>
<tr>
<td><strong>Physical scale [µm/pixel]</strong></td>
<td>70</td>
<td>115</td>
</tr>
<tr>
<td><strong>Magnification [-]</strong></td>
<td>0.39</td>
<td>0.24</td>
</tr>
</tbody>
</table>

### 3.2 Fuels

The fuels used in all experiments in this work were either propane, DME, or propane-DME blends. Oberon Fuels supplied the DME and propane-DME blends while pure propane was purchased from Airgas. Three propane-DME blends were used in this work: 90%-10%, 80%-20%, and 70%-30% propane-DME by mass, hereafter referred to as 90-10, 80-20, and 70-30, respectively. Table 3.4 provides important properties of propane and DME compared to #2 diesel fuel.
Table 3.4: Important properties of propane, DME, diesel.

<table>
<thead>
<tr>
<th></th>
<th>Units</th>
<th>#2 Diesel</th>
<th>Propane</th>
<th>DME</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td></td>
<td>C\textsubscript{n}H\textsubscript{1.87n}</td>
<td>C\textsubscript{3}H\textsubscript{8}</td>
<td>C\textsubscript{2}H\textsubscript{6}O</td>
</tr>
<tr>
<td>Liquid density (T = 20°C)</td>
<td>kg/m\textsuperscript{3}</td>
<td>820-845</td>
<td>499.8</td>
<td>669.4</td>
</tr>
<tr>
<td>Boiling point (P = 1 atm)</td>
<td>°C</td>
<td>180-360°C</td>
<td>-42.1</td>
<td>-24.9</td>
</tr>
<tr>
<td>Cetane number</td>
<td></td>
<td>40-55</td>
<td>-</td>
<td>55-60</td>
</tr>
<tr>
<td>LHV</td>
<td>MJ/kg</td>
<td>43</td>
<td>46.34</td>
<td>28.9</td>
</tr>
<tr>
<td>Volumetric energy density</td>
<td>MJ/L</td>
<td>35.2-36.3</td>
<td>23.2</td>
<td>19.3</td>
</tr>
<tr>
<td>Stoichiometric AFR</td>
<td></td>
<td>14.6</td>
<td>15.6</td>
<td>9.0</td>
</tr>
<tr>
<td>Kinematic viscosity (T = 25°C)</td>
<td>cSt</td>
<td>1.9-4.1</td>
<td>0.20</td>
<td>0.18</td>
</tr>
</tbody>
</table>

The fuel system on the CFR engine used to carry out the ASTM D613 cetane number standard test is not compatible with liquefied gases, and therefore limited cetane number information is available for both propane and DME. Since DME has been a fuel of interest for CIDI engines for some time, there has been an effort to estimate the cetane number based on engine data \cite{7}, which is what is presented in Table 3.4. Propane is much less reactive than DME under CIDI engine conditions and has not been researched as a potential CIDI fuel as heavily as DME, so the cetane number is not well defined.

DME has a significantly lower LHV than both propane and diesel, which have comparable LHVs, but since propane is less dense than DME, which is less dense than diesel, the volumetric energy density of both propane and DME is significantly lower than that of diesel. Consumers typically purchase fuel on a volumetric basis, so volumetric energy density is an important consideration when comparing fuels. Since DME is an oxygenated fuel, the stoichiometric air-fuel ratio (AFR) is lower than both diesel and propane.

The viscosity of propane and DME are around an order of magnitude less than that of diesel, which can be detrimental to fuel system components where relative motion is present because the significant decrease in viscosity brings lubrication towards the boundary lubrication regime \cite{53} where loads are supported solely by asperity contact. Also, propane and DME are much more compressible than diesel at CIDI-relevant injection pressures leading to increased pumping work and reduced injection pressure capability. Figure 3.17 shows the isothermal bulk modulus (\(K_v\)) of propane, DME, and diesel \cite{54} as a function of pressure.
from 100 to 2000 bar. The isothermal compressibility of a fluid is defined by

\[ K_v = -V \frac{\partial P}{\partial V} \bigg|_T \]  

(3.2)

where \( V \) is the initial volume of the fluid.

Figure 3.17: Isothermal bulk modulus of propane, DME, and diesel as a function of pressure.

Figure 3.17: Isothermal bulk modulus of propane, DME, and diesel as a function of pressure.
Chapter 4

Non-reacting jet measurements

4.1 Introduction

The physical evolution of fuel jets in CIDI engines plays a crucial role in the combustion process. At high pressure, liquid fuel undergoes several key stages before combustion reactions occur, including atomization, liquid heating, and eventual vaporization. Injecting fuel into a non-reacting environment of pure nitrogen gas provides an opportunity to study and isolate the physical processes involved in jet development, free from the complicating effects of combustion reactions. This chapter discusses the behavior of non-reacting propane, DME, and propane-DME fuel jets injected at high pressure at varying SOI in-cylinder densities of 7.5, 15, and 22.5 kg/m$^3$. These densities represent an early pilot, a late pilot, and a main injection on a typical light-duty CIDI engine.

Schlieren imaging was utilized in this study. This imaging method captures density gradients, representing them as intensity variations in an image. An image processing routine, discussed in Section 3.1.4, was employed to calculate penetration and spray dispersion angle to quantify the injection characteristics.
4.2 Experimental details

4.2.1 Operating parameters

Propane, DME, and all three propane-DME blends (90-10, 80-20, and 70-30) were injected into pure nitrogen gas at SOI in-cylinder densities of 7.5, 15, and 22.5 kg/m$^3$, representative of an early pilot, late pilot, and main injection, respectively. Intake flow rate and injection timings were adjusted to attain these specific densities, as outlined in Table 4.1.

Table 4.1: Important parameters for non-reacting SOI in-cylinder density sweep.

<table>
<thead>
<tr>
<th>SOI density [kg/m$^3$]</th>
<th>7.5</th>
<th>15</th>
<th>22.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake flow rate [g/s]</td>
<td>4.52</td>
<td>5.51</td>
<td>8.10</td>
</tr>
<tr>
<td>Intake pressure [bar]</td>
<td>1.04</td>
<td>1.26</td>
<td>1.84</td>
</tr>
<tr>
<td>SOI [CAD]</td>
<td>-28.75</td>
<td>-9</td>
<td>-4.75</td>
</tr>
<tr>
<td>Intake temperature [°C]</td>
<td></td>
<td>28</td>
<td></td>
</tr>
<tr>
<td>Rail pressure [bar]</td>
<td></td>
<td>500</td>
<td></td>
</tr>
<tr>
<td>Injection duration [ms]</td>
<td></td>
<td>0.5</td>
<td></td>
</tr>
</tbody>
</table>

Figure 4.1 displays the motored pressure and isentropic temperature with SOI timings for the different SOI in-cylinder densities represented by the vertical lines. Since intake temperature was held constant for all SOI in-cylinder densities, the isentropic temperature did not increase as significantly as the cylinder pressure as SOI density increased.

Figure 4.2, along with the data presented in Figure 4.1, shows that all of the sprays were vaporizing. The SOI isentropic temperatures were much higher than the saturation temperature of propane or DME at the SOI cylinder pressures. It is important to confirm that the sprays were vaporizing as vaporization has been shown to slightly impact spray penetration [16].

It is important to note that, as stated in [16], jets do not typically reach a quasi-steady state until approximately 0.5 ms after SOI (aSOI). In the optical engine setup, injection durations are constrained by the moment the jet collides with the piston bowl wall, occurring roughly between 0.4 and 0.6 ms aSOI, depending on in-cylinder density. Once the jet makes contact with the piston bowl, its penetration and dispersion characteristics become unreliable.
Figure 4.1: Cycle-averaged pressure and isentropic temperature as a function of CAD for all SOI in-cylinder densities. The vertical dash-dotted lines represent SOI timings for the different in-cylinder densities.

Figure 4.2: Saturation temperature as a function of pressure for propane and DME from 100 kPa up to the critical pressure.
as the fuel plume rapidly spreads along the periphery of the piston bowl, which is why injection durations weren’t extended beyond 0.5 ms for these experiments.

The choked flow orifice rack employed to measure the intake flow rate underwent calibration using air. However, the indicated flow rate does not factor in the density variations associated with using pure nitrogen. It is imperative to account for these differences before operations begin to ensure knowledge of the intake nitrogen flow rate. Initially, the engine underwent "scoping" with air. This involved motoring the engine at varying intake air flow rates to establish the mass flow rate and injection timings necessary to achieve the specified SOI in-cylinder densities. Once the correct mass flow rates were determined using air, the required indicated air flow rate for pure nitrogen was computed based on the choked orifice mass flow relationship,

\[ \dot{m} = C_d A \sqrt{2 \rho (P_1 - P^*)} \]  \hspace{1cm} (4.1)

where \( C_d \) is the orifice discharge coefficient, \( A \) is the area of the orifice, \( \rho \) is the fluid density upstream of the orifice, \( P_1 \) is the upstream pressure, and \( P^* \) is the critical pressure given by

\[ \frac{P^*}{P_0} = \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \]  \hspace{1cm} (4.2)

where \( P_0 \) is the inlet stagnation pressure and \( \gamma \) is the specific heat ratio. Assuming \( \gamma_{\text{air}} \approx \gamma_{\text{N}_2} \), the ratio of nitrogen mass flow to air mass flow at the same upstream pressure can be simplified to

\[ \frac{\dot{m}_{\text{N}_2}}{\dot{m}_{\text{air}}} = \sqrt{\frac{\rho_{\text{N}_2}}{\rho_{\text{air}}}} \]  \hspace{1cm} (4.3)

which was used to calculate the required indicated flow rate to match nitrogen and air mass flow.
4.3 Results and discussion

4.3.1 Penetration

Figure 4.3 displays penetration results for 20 injections of propane injected at a SOI in-cylinder density of 7.5 kg/m$^3$. Penetration cannot be calculated early in the injection when the jet has not penetrated past the injector hole boundary (4.5 mm). The penetration rate decreases throughout the injection process as expected due to air entrainment and conservation of momentum. Once the jet reaches the piston bowl wall, corresponding to a penetration of approximately 26.7 mm, the penetration is no longer meaningful as it remains constant for the rest of the injection.

![Figure 4.3: Jet penetration vs. time aSOI for a propane jet at an SOI in-cylinder density of 7.5 kg/m$^3$. The dotted red lines are individual injections, and the solid black line is the mean penetration curve. The shaded gray region corresponds to ±1 standard deviation about the mean.](image)

In Figure 4.4, the penetration results for all fuels at differing SOI in-cylinder densities are shown. The influence of fuel composition on penetration is less significant than the impact of ambient gas density. This observation aligns with previous studies on vaporizing jets, as penetration has been shown to have a weak dependence on the ratio of ambient gas
density ($\rho_a$) to fuel density ($\rho_f$) \cite{16,18}. The increase in fuel density is modest, with DME blend percentages ranging from 0% to 100% resulting in only a 1.34-fold increase, whereas the ambient density increases by a factor of three, rising from 7.5 kg/m$^3$ to 22.5 kg/m$^3$. The error bands on the propane and DME penetration curves represent the 95% confidence interval on the mean and do not overlap for most of the injection at the 7.5 kg/m$^3$ SOI density. The propane penetrates significantly faster than the DME, which does not agree with trends observed in the literature. For the 15 kg/m$^3$ in-cylinder density, propane penetrates slower than DME, and the difference becomes significant near the end of the injection. At an SOI density of 22.5 kg/m$^3$, there is no significant difference between the penetration for any of the fuels.

### 4.3.2 Dispersion

Figure 4.5 displays dispersion half-angle results for 20 injections of propane injected at a SOI in-cylinder density of 7.5 kg/m$^3$. Dispersion cannot be calculated until the jet has penetrated about twice the distance of the injector hole radius (9 mm) because only the upstream half of the jet is used to calculate dispersion. The average dispersion half-angle increases from 0.18 radians at 200 $\mu s$ aSOI to 0.20 radians at 400 $\mu s$ when the jet contacts the piston bowl wall evidenced by the average penetration curve in Figure 4.3. This increase in dispersion is likely due to the processing method using frame-differencing to build the binarized jet which only allows for area to be added to the jet and not removed. Another explanation for the increase in dispersion angle is the increasing ambient gas density as it is compressed by the piston.

Figure 4.6 shows cycle-averaged dispersion results for all fuels at the different SOI densities and solely propane and DME results with a 95% confidence interval about the mean. Previous spray studies have found the tangent of the dispersion half-angle to be proportional to $(\rho_a/\rho_f)^n$ where $n = 0.19$ \cite{16}. At constant SOI density, the difference in dispersion half-angle between the blends is small and hard to decipher from Figure 4.6 which is expected
Figure 4.4: Cycle-averaged penetration vs. time aSOI for all five fuels across different SOI in-cylinder densities. The legend numbers correspond to the composition of the propane-DME blend, with the first indicating the percentage of propane by mass and the second indicating the percentage of DME by mass. Error bands for pure propane and DME represent the 95% confidence interval on the mean.
because the density of the 30% DME blend is only 5% greater than that of the 10% DME blend which would only explain a decrease in the tangent of the dispersion half-angle of 1% based on the relationship from [16]. While the difference in density between the blends is small, DME is 34% more dense than propane, which is large enough to cause a noticeable difference in dispersion angle. Since propane is less dense than DME, the dispersion angle should be larger, which is evident at the 15 kg/m$^3$ and 22.5 kg/m$^3$ SOI densities. At the 7.5 kg/m$^3$ SOI density, the opposite trend is true at the beginning of the dispersion trace, but by the time the jet hits the wall around 400 $\mu$s, the dispersion angles are nearly equal.

Contrary to previous studies and intuition, the dispersion angle at the 7.5 kg/m$^3$ SOI density is not consistently less than the dispersion angle at 15 kg/m$^3$ density. Analyzing the single-shot images like the image in Figure 4.7 showed that at the low SOI density, there was sometimes a significant dispersed vapor region at the head of the fuel jet that was not present at higher densities.

Ensemble averaging the images before processing smoothed out some of the cycle-to-
Figure 4.6: Cycle-averaged dispersion vs. time aSOI for all five fuels (left) across different SOI in-cylinder densities and propane/DME only results with confidence intervals about the mean (right). The legend numbers correspond to the composition of the propane-DME blend, with the first number indicating the percentage of propane by mass and the second indicating the percentage of DME by mass.
cycle variability and significantly changed the dispersion results, as shown in Figure 4.8. The dispersion angle was reduced significantly, and the shape of the dispersion versus time aSOI curve more closely resembled results from [16].

Figure 4.8: Average dispersion calculated from single-shot images (black) and dispersion calculated from ensemble-averaging (blue) for propane blend at the 7.5 kg/m$^3$ SOI density.

Figure 4.9 shows the dispersion half-angle calculated using the ensemble-averaged images for propane and DME at all in-cylinder SOI densities. For propane, the dispersion angle follows the expected trend: increasing dispersion with in-cylinder density. The same is not true for DME, where the dispersion angle at 400 µs aSOI at an in-cylinder density of 7.5 kg/m$^3$ is greater than at the 15 kg/m$^3$ in-cylinder density. The impact of fuel density on the dispersion angle follows the expected trend for the 15 kg/m$^3$ and 22.5 kg/m$^3$ in-cylinder...
densities where the less dense fuel (propane) has a larger dispersion angle. The opposite is true for the 7.5 kg/m$^3$ in-cylinder density, which aligns with findings illustrated in Figure 4.6. Although ensemble-averaging removed some of the cycle-to-cycle variability from the images and yielded dispersion angles more representative of quasi steady-state fuel jets, the trends in dispersion angle remained the same.

![Figure 4.9: Dispersion half-angle calculated from ensemble-averaged images for propane (solid lines) and DME (dashed lines) at all SOI in-cylinder densities.]

4.4 Conclusions

Schlieren images of non-reacting, vaporizing fuel jets were acquired using propane, DME, and three different propane-DME blends at three different SOI in-cylinder densities. Penetration results indicated that as SOI in-cylinder density increased, the penetration rate decreased, as evidenced by increased time aSOI to reach the piston bowl wall with increasing SOI density. The penetration rate also decreased with time aSOI due to air entrainment and conservation of momentum, leading to a reduction in velocity. Dispersion results were not significantly different between blends, but there was a noticeable difference in dispersion
between propane and DME due to the larger difference in density. While it was expected for
the lowest SOI density to yield the smallest dispersion angle, this was not observed, likely
due to the transient and turbulent nature of the jets, often creating a dispersed vapor region
at the head of the jet at low ambient densities. Utilizing ensemble averaging of the images
smoothed out some of the transience, and the dispersion angle versus time aSOI curve more
closely resembled data from previous studies, but the transient behavior is real and was
captured in the single-shot images.
Chapter 5

Single-injection experiments

5.1 Introduction

Modern CIDI engines have evolved to incorporate multiple injections per cycle. This advanced injection strategy serves various purposes, including the precise control of combustion phasing, reduction of engine noise, moderation of exhaust temperatures, and compliance with stringent emissions regulations. However, in the context of unconventional fuels such as propane and propane-DME blends, which are not commonly used in CIDI engines, simplifying the combustion process becomes necessary when studying them in an optically accessible engine. This simplification often involves employing a single-injection strategy.

This chapter delves into the outcomes of three separate optical engine studies of single injections of propane, DME, and propane-DME blends. These studies were designed to explore the combustion characteristics of these fuels under simplified conditions. The first study focused on varying the injection timing, a critical parameter influencing combustion behavior. The insights gained from this study were then applied in the second investigation, which explored the effects of intake temperature on combustion. Finally, the third study involved sweeping through different levels of injected fuel energy.

Notably, the sequence of these studies was planned to build upon one another. The results
from the injection timing study informed the intake temperature study and the findings from the intake temperature study influenced the approach taken in the fuel energy study. This sequential approach ensured a thorough investigation of the combustion characteristics of these fuels under simplified conditions.

5.2 Experimental details

5.2.1 Operating parameters

Injection timing sweep

Intake thermodynamic conditions, rail pressure, and commanded injection duration were held constant for the injection timing sweep and are displayed in Table 5.1.

Table 5.1: Operating parameters that were held constant for all fuels.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>rpm</td>
<td>1200</td>
</tr>
<tr>
<td>Intake pressure</td>
<td>bar</td>
<td>2.99</td>
</tr>
<tr>
<td>Intake temperature</td>
<td>C</td>
<td>150</td>
</tr>
<tr>
<td>Rail pressure</td>
<td>bar</td>
<td>500</td>
</tr>
<tr>
<td>Commanded injection duration</td>
<td>ms</td>
<td>0.4</td>
</tr>
</tbody>
</table>

The relatively low compression ratio of the optical engine (12.0:1) coupled with the low reactivity of propane and propane-DME blends necessitates elevated intake temperatures and pressures to achieve reliable combustion. A short injection duration was employed to reduce potentially high maximum pressure rise rates (MPRR) due to premixed combustion associated with the long ignition delays of low-reactivity fuels. Under the conditions displayed in Table 5.1, the average peak motored in-cylinder pressure was 79 bar, and the maximum isentropic temperature was around 1060 K. Figure 5.1 shows the average in-cylinder motored pressure and isentropic temperature traces as a function of crank-angle.

Table 5.2 displays the swept start-of-injection (SOI) timings. A 10 CAD range was swept in 2.5 CAD increments for all fuels, yielding five runs per fuel. The schlieren images
Figure 5.1: Average in-cylinder motored pressure and temperature for the injection timing sweep. The shaded region corresponds to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.
determined SOI to succeed the injection command signal timing by 0.5 CAD. SOI timings for propane and the propane-DME blends were selected to provide adequate time for the low-reactivity fuels to ignite, while the SOI timings for DME were all retarded from the propane and propane-DME blend timings by 7.5 CAD due to the high-reactivity of DME.

Table 5.2: SOI timings for the different fuels.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>SOI [CAD]</th>
</tr>
</thead>
<tbody>
<tr>
<td>DME</td>
<td>-0.75, -3.25, -5.75, -8.25, -10.75</td>
</tr>
</tbody>
</table>

Intake temperature sweep

Intake temperature was swept from 120-200°C in 20°C increments using the same 0.4 ms injection duration used in the injection timing sweep. Table 5.3 displays the intake thermodynamic conditions for the intake temperature sweep.

<table>
<thead>
<tr>
<th>Intake Temperature [C]</th>
<th>Intake Pressure [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>120</td>
<td>2.18</td>
</tr>
<tr>
<td>140</td>
<td>2.27</td>
</tr>
<tr>
<td>160</td>
<td>2.35</td>
</tr>
<tr>
<td>180</td>
<td>2.44</td>
</tr>
<tr>
<td>200</td>
<td>2.51</td>
</tr>
</tbody>
</table>

Table 5.3: Intake thermodynamic conditions for intake temperature sweep.

It is important to note that the intake air mass flow rate was constant at 7.85 g/s for all intake temperatures which is why intake pressure increases with intake temperature. Figure 5.2 shows the maximum average isentropic compressed gas temperature and the equivalent compression ratio (CR) as a function of intake temperature. The equivalent CR was calculated as required to achieve the corresponding isentropic compressed gas temperature, assuming an intake pressure of 1 atm and a temperature of 40 40°C. The equivalent CR was solved iteratively by first setting the specific entropies at the intake state and the compressed state equal to calculate the compressed state pressure. Then, the densities at the intake and compressed state were calculated using the respective pressures and temperatures, and the
CR was taken as the ratio of the compressed state density to the intake state density. All properties were calculated using an equation of state (EOS) for air developed by Lemmon et al. [55] that is built-in to the general equation solving-program Engineering Equation Solver (EES) [56].

The in-cylinder intake temperature was chosen based on in-cylinder, optical engine temperature measurements performed by Bürkle et al. [57] using tunable diode laser absorption spectroscopy where the authors found that in-cylinder temperature during the intake stroke lies in between the temperature upstream of the intake valve and the temperature of the cylinder head. Room temperature in the optical engine laboratory ranges from 20-25°C, and the head temperature was assumed to be equal to the coolant temperature (50°C). Hence, the in-cylinder intake temperature was set to 40°C.

Figure 5.2: Maximum cycle-averaged motored in-cylinder temperatures as a function of intake temperature and simulated CR based on compressed gas temperature. Error bars correspond to the 95% confidence interval about the mean.

Over the swept intake temperature range, maximum in-cylinder temperature trended
linearly with intake temperature from a minimum of 1005 K at an intake temperature of 120°C to a maximum of 1115 K at an intake temperature of 200°C. Injection timings for the intake temperature sweep were determined from the injection timing sweep as the timings that yielded the highest gross indicated mean effective pressure (IMEP\textsubscript{g}). Table 5.4 shows the SOI timings used for the intake temperature sweep. The injection timing sweep experiment will be discussed in detail later in this chapter.

Table 5.4: SOI timings used in the intake temperature sweep.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>SOI [CAD]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propane</td>
<td>-13.25</td>
</tr>
<tr>
<td>90-10</td>
<td>-10.75</td>
</tr>
<tr>
<td>80-20</td>
<td>-10.75</td>
</tr>
<tr>
<td>70-30</td>
<td>-8.25</td>
</tr>
<tr>
<td>DME</td>
<td>-3.25</td>
</tr>
</tbody>
</table>

**Injected fuel energy sweep**

The injection timings were kept constant during the fuel energy sweep, following the settings from the intake temperature sweep. Additionally, the intake conditions were set to match those of the 180°C intake temperature condition used in the intake temperature sweep. This specific intake temperature was selected because it represented the lowest temperature at which there was no significant decrease in cumulative heat release compared to the 200°C intake temperature across all fuels. This choice was informed by a detailed heat release analysis, which will be further elaborated upon in subsequent sections of this chapter.

The injected fuel energy spanned a range from 275 to 675 J, incremented in 100 J steps. The lower end of this spectrum represented a 0.4 ms injection of an 80-20 blend at 500 bar, while the upper end was representative of a 1.5 ms injection of DME at 500 bar. The relationship between injection durations and all fuels' corresponding injected fuel energy is detailed in Table 5.5. Due to DME's lower LHV than propane, an increase in the DME blend percentage necessitates a longer injection duration to maintain a constant injected energy, as evidenced by the trends in Table 5.5.
Table 5.5: Injected fuel energy and commanded injection duration for all fuels.

<table>
<thead>
<tr>
<th>Injection duration [ms]</th>
<th>Fuel Energy [J]</th>
<th>275</th>
<th>375</th>
<th>475</th>
<th>575</th>
<th>675</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propane</td>
<td>0.38</td>
<td>0.51</td>
<td>0.64</td>
<td>0.79</td>
<td>0.98</td>
<td></td>
</tr>
<tr>
<td>90-10</td>
<td>0.39</td>
<td>0.52</td>
<td>0.65</td>
<td>0.82</td>
<td>1.02</td>
<td></td>
</tr>
<tr>
<td>80-20</td>
<td>0.40</td>
<td>0.54</td>
<td>0.68</td>
<td>0.85</td>
<td>1.05</td>
<td></td>
</tr>
<tr>
<td>70-30</td>
<td>0.41</td>
<td>0.55</td>
<td>0.69</td>
<td>0.88</td>
<td>1.09</td>
<td></td>
</tr>
<tr>
<td>DME</td>
<td>0.52</td>
<td>0.72</td>
<td>0.98</td>
<td>1.23</td>
<td>1.5</td>
<td></td>
</tr>
</tbody>
</table>

5.3 Results and discussion

5.3.1 Injection timing sweep

The pressure versus crank angle data obtained from the injection timing sweep were utilized to calculate the IMEP\textsubscript{g} for each fired cycle. The SOI timing that resulted in the highest cycle-averaged IMEP\textsubscript{g} was chosen for use in the subsequent experiments. Table 5.5 displays the injection timings that produced the highest IMEP\textsubscript{g}, along with the corresponding IMEP\textsubscript{g} values for each type of fuel and the injected fuel energy.

Table 5.6: SOI, indicated load, and injected fuel energy for all fuels.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>SOI [CAD]</th>
<th>IMEP\textsubscript{g} [bar]</th>
<th>E\textsubscript{fuel} [J]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propane</td>
<td>-13.25</td>
<td>0.83</td>
<td>280</td>
</tr>
<tr>
<td>90-10</td>
<td>-10.75</td>
<td>0.83</td>
<td>272</td>
</tr>
<tr>
<td>80-20</td>
<td>-10.75</td>
<td>0.71</td>
<td>264</td>
</tr>
<tr>
<td>70-30</td>
<td>-8.25</td>
<td>0.64</td>
<td>257</td>
</tr>
<tr>
<td>DME</td>
<td>-3.25</td>
<td>0.41</td>
<td>197</td>
</tr>
</tbody>
</table>

As the percentage of DME blend increases, the IMEP\textsubscript{g} decreases primarily due to the LHV reduction. DME has an LHV of 28.9 MJ/kg compared to propane’s 46.34 MJ/kg. Although an increase in density partially offsets the decrease in LHV as the DME blend percentage rises, resulting in a slight increase in injected mass, the overall reduction in LHV has a more significant impact. Increasing the DME blend percentage reduces the ignition delay, requiring the adjustment of injection timings to maximize IMEP\textsubscript{g}. The additional decrease in IMEP\textsubscript{g} beyond what is expected from the decrease in fuel energy content for the 80-20 and 70-30 blends is not well understood and could be due to a number of reasons. One
possible explanation is that the difference in ambient conditions between the engine and the mass measurement vessel led to variations in injected fuel energy.

As the DME blend percentage increased, pressure rise rate and oscillations increased. This trend is evident when examining the single-cycle raw pressure traces in Figure 5.3 where the 70-30 and 80-20 blends demonstrate significantly higher pressure rise rates and pressure oscillations compared to the 90-10 blend.

Figure 5.3: Raw single-cycle pressure traces for the three blends at the maximum IMEP₇ timings.

SOC from the raw pressure signal was used to calculate the ignition delay (ID). Figure 5.4 displays the ID as a function of SOI for propane and all blends. It is apparent from Figure 5.4 that, generally, the ID decreases with both an increasing percentage of DME in the blend and retarding SOI timing. However, there is an exception with propane, where the ID appears to increase slightly when SOI is retarded from -10.75 to -8.25 CAD. This phenomenon is due to propane’s low reactivity, long ignition delay, and the proximity of SOI to the expansion stroke, causing ignition to occur when gas temperatures and reaction rates
begin to decrease rapidly. The ID for DME is less than 500 $\mu$s for all SOI timings and does not vary significantly due to its high reactivity and injection timings close to TDC.

Figure 5.4: Pressure-based ID vs. SOI for all fuels. Error bars correspond to a 95% confidence interval about the mean.

Figure 5.5 depicts the HRR results corresponding to the timings that yielded the highest indicated mean effective pressure IMEP$_g$. As the percentage of DME increases from 0% to 30%, the heat release event becomes more abrupt, which is attributed to the decrease in ignition delay and heightened reactivity of the fuel. In contrast, for pure DME, there is a distinct premixed spike in the heat release, followed by a brief phase of mixing-controlled combustion. This trend observed with the blends aligns with the findings in [58], where it was noted that the burn period increased with ignition delay for "mainly premixed" diesel combustion which encompasses the propane and propane-DME blends as SOC does not occur until after end-of-injection (EOI) for propane, 90-10, and 80-20, and nearly coincides with EOI for the 70-30 blend.
The trend of increased premixing, attributed to longer ignition delays as the DME blend percentage decreases, is clearly illustrated in the OH* chemiluminescence images depicted in Figure 5.6. OH* chemiluminescence signal typically increases with higher equivalence ratios in lean and stoichiometric regions [59]. Given that the combustion process is predominantly premixed for propane and propane-DME blends, it logically follows that the OH* chemiluminescence signal should rise as the ignition delay decreases.

For propane and the 90-10 blend, the OH* chemiluminescence signal is notably faint and challenging to discern without aid. However, as the DME blend percentage increases from 10% to 20% and then from 20% to 30%, there is a marked increase in the OH* chemiluminescence signal. This can be attributed to the local equivalence ratio becoming richer due to the shorter ignition delay.
Figure 5.6: OH* images from injection timing sweep for all fuels.
5.3.2 Intake temperature sweep

The intake temperature sweep results showed that all fuels reliably ignited at an intake temperature $\geq 180$ °C. Below 180 °C, combustion efficiency notably decreased for propane and the 90-10 blend. Figure 5.7 illustrates this trend. Because of the reduced combustion efficiency observed at 160 °C, the fuel energy sweep was conducted at the 180 °C intake temperature condition, as determined from the intake temperature sweep.

![Figure 5.7: Intake temperature sweep cycle-averaged cumulative heat release for propane (left) and the 90-10 blend (right). The shaded region corresponds to a 95% confidence interval about the mean due to cycle-to-cycle variability.](image)

In addition to calculating SOC from pressure, OH* chemiluminescence images were used to estimate SOC for the intake temperature sweep. The images were processed to determine SOC utilizing the average intensity threshold method outlined in section 5.2.

Start of combustion calculated from OH* chemiluminescence images typically precedes pressure-based SOC for a few reasons. Firstly, the pressure transducer on the optical engine is positioned on the cylinder wall, and the time required for pressure waves generated due to ignition to reach the transducer is non-negligible. This time will vary depending on the ignition location but is estimated to be 50-100 µs. Secondly, the average intensity
threshold method is very sensitive and is triggered by small increases in signal level. Lastly, a significant amount of heat release is needed to increase in-cylinder pressure enough for detection. The intake temperature sweep results reveal that the combination of these effects leads to a reduction of approximately 200 µs (1.4 CAD) in ID when calculated using OH* chemiluminescence instead of pressure.

Figure 5.8 illustrates the difference between OH* chemiluminescence and pressure-based ID and demonstrates the strong decrease in ID with increasing temperature for all fuels, consistent with expectations. The results indicate that despite reaching the highest intake temperature, ID decreases rapidly for pure propane and the propane-DME blends. In contrast, the ID for DME remains relatively stable, suggesting that at these temperatures, the ID of DME is primarily limited by fuel-air mixing, whereas the ID of propane and the blends is predominantly governed by chemical kinetics.

Figure 5.8: Pressure-based ID and OH* chemiluminescence-based ID vs. intake temperatures for all fuels. Error bars correspond to a 95% confidence interval about the mean. SOI timing information can be found in Table 5.6.
5.3.3 Injected fuel energy sweep

The SOI timings remained consistent with the intake temperature sweep, except for the 80-20 and 90-10 fuel blends at the highest injected fuel energies. In these cases, the SOI was retarded to -8.75 and -6.75 CAD, respectively, to limit the maximum pressure rise rate.

It is apparent from the cycle-averaged images from the highest fuel energy case shown in Figure 5.9 that all fuels except 70-30 and DME ignite after the EOI. This is indicated by the OH* chemiluminescence signal occurring only on the periphery of the piston bowl, with no evident jet structure present in the images. These observations suggest that the combustion is not mixing-controlled. For the 70-30 blend, SOC happened just before EOI. Autoignition occurred in the partially premixed region near the bowl’s periphery. This was followed by a brief period of mixing-controlled combustion near TDC, as indicated by the jet structures visible in the OH* chemiluminescence images. The DME case ignited rapidly, with only a small portion of the combustion event being premixed. Following this initial premixed combustion, the process transitioned immediately to a mixing-controlled combustion, which lasted until the injected fuel was consumed (shortly after EOI).
Figure 5.9: Cycle-averaged OH* chemiluminescence images at all fuels’ highest injected fuel energy. The corresponding CAD is in the upper left-hand corner of each frame.
5.4 Conclusions

As expected, experiments revealed that injection timing significantly influences combustion behavior, with the optimal timing varying depending on the fuel blend. As the DME blend percentage increased, injection timing was retarded from -13.25 CAD for pure propane to -8.25 CAD for the 70-30 blend to achieve the highest indicated efficiency. Intake temperature also plays a crucial role, with temperatures below 180 °C leading to reduced combustion efficiency for propane and propane DME blends. DME, on the other hand, demonstrated rapid ignition and consistent performance across a range of conditions. High-pressure rise rates due to abrupt pre-mixed combustion near TDC are not desirable, and a pilot injection is needed to increase the temperature and pressure before the main injection to decrease the ignition delay of the main injection and transition towards MCCI.

Intake heating on automotive engines is challenging to implement due to transient intake flow rates and environmental conditions. Increasing the compression ratio has a similar effect but can lead to high cylinder pressures that can damage engine components. Utilizing an ignition assistant like a glow plug, plasma source, or active pre-chamber can also improve the combustion reliability and efficiency of low-reactivity fuels like propane without substantially changing an engine’s architecture.
Chapter 6

Split-injection autoignition experiments

6.1 Introduction

In modern CI engines, multiple injection strategies are commonly employed, with some engines using 5 fuel injections per cycle [60]. These strategies are used to manage combustion noise, emissions, and exhaust gas temperature (EGT). Injection events are typically classified as pilot, main, or post injections based on their timing and the amount of fuel delivered. Pilot injections deliver a small amount of fuel before the main injection event, while post injections occur after the main injection. The practice of using pilot injections to mitigate combustion-related noise in diesel engines dates back to at least 1939 [61]. By utilizing pilot injections, a prolonged chemical heat release event can be achieved, limiting the maximum heat release rate and reducing pressure rise rates. The pre-mixed combustion resulting from some of the pilot injection leads to an increase in in-cylinder gas temperature and pressure, thereby reducing the ignition delay of the main injection and increasing the mixing-controlled portion of the heat release [62]. For low-reactivity fuels that resist auto-ignition, the pilot fuel quantity must be sufficient to elevate in-cylinder temperatures and pressures enough for the main injection to shift towards mixing-controlled combustion. This necessitates larger pilot injections that can equal or exceed the mass of the main injection. The pilot-main
strategy significantly depends on the timing of the pilot injection, with the ideal timing varying with fuel reactivity. This section explores the impact of pilot injection timing by maintaining a constant main injection timing of -5 CAD while varying the pilot timing from -50 to -20 CAD.

6.2 Experimental details

6.2.1 Operating parameters

Intake conditions were held constant for all runs at the same conditions used for the single injection fuel energy sweep detailed in Section 5.2.1. The pilot and main injection commanded injection durations were split evenly at 0.4 ms each, and injection pressure was held constant at 500 bar. Table 6.1 displays important engine and injection parameters.

Table 6.1: Operating parameters that were held constant for all fuels. *The reported timings correspond to the start of fuel flow into the cylinder, not the injector command timing, which precedes SOI by 0.5 CAD.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake pressure</td>
<td>bar</td>
<td>2.45</td>
</tr>
<tr>
<td>Intake temperature</td>
<td>C</td>
<td>180</td>
</tr>
<tr>
<td>Rail pressure</td>
<td>bar</td>
<td>500</td>
</tr>
<tr>
<td>Commanded injection duration (pilot/main)</td>
<td>ms</td>
<td>0.4/0.4</td>
</tr>
<tr>
<td>Pilot injection timing (SOI)*</td>
<td>CAD</td>
<td>Varied from -50 to -20</td>
</tr>
<tr>
<td>Main injection timing (SOI)*</td>
<td>CAD</td>
<td>-5</td>
</tr>
</tbody>
</table>

Prior to conducting the timing sweep on the engine, the injection mass for each timing and fuel type was measured using the instrumented vessel and method detailed in Section 5.2. A notable advantage of employing a direct-acting piezoelectric injector is its fast needle motion, resulting in injected masses that are more consistent and less influenced by dwell ($t_d$)—the time between two consecutive injections—compared to a solenoid injector. In this study, dwell is defined as the duration between the SOI of the first injection and the SOI of the second injection. Figure 6.1 illustrates the relationship between injected mass, dwell,
and pilot SOI. Generally, injection mass tends to increase with dwell, with a few exceptions noted for propane and the 90-10 blend. However, the uncertainty bands almost entirely encompass the range of injection masses across varying dwell times for all fuels.

![Figure 6.1: Total injection mass vs injection dwell for all fuels.](image)

Dwell times varied between less than 2 ms and nearly 6 ms, depending on the timing of the pilot injection. Table 6.2 provides the specific dwell times corresponding to the swept pilot SOI timings.

Table 6.2: Injection dwell and the corresponding pilot SOI. Main injection SOI was held constant at -5 CAD

<table>
<thead>
<tr>
<th>Pilot SOI [CAD]</th>
<th>Dwell [ms]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-20</td>
<td>1.68</td>
</tr>
<tr>
<td>-27.5</td>
<td>2.73</td>
</tr>
<tr>
<td>-35</td>
<td>3.77</td>
</tr>
<tr>
<td>-42.5</td>
<td>4.81</td>
</tr>
<tr>
<td>-50</td>
<td>5.85</td>
</tr>
</tbody>
</table>

As the injection mass showed only slight variations with dwell, the pilot mass fraction
remained nearly constant across all dwells, ranging from 51% to 54%. The injected fuel energies spanned from 485 J for the shortest dwell using the 70-30 blend to 540 J for the longest dwell using propane.

The in-cylinder thermodynamic conditions at the time of SOI varied significantly due to the rapid compression of the gas, leading to notable changes in pressure and temperature with crank angle. Figure 6.2 illustrates the cycle-averaged pressure and isentropic temperature for motored cycles, with vertical lines indicating the pilot SOIs and the main SOI. It is important to consider that fuel injection influences these thermodynamic conditions through the evaporative cooling process. Additionally, in some cases, the start of combustion SOC preceded the main SOI, resulting in significant changes to the thermodynamic conditions at the main SOI timing. It is worth noting that Figure 6.2 does not account for these changes and serves only as a reference to illustrate the motored thermodynamic conditions for the pilot timing sweep.
Figure 6.2: Average in-cylinder motored pressure and temperature with the pilot and main SOI timings overlaid. The shaded region corresponds to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.
6.3 Results and discussion

6.3.1 Ignition delays

As the pilot timing was advanced, the ID increased as anticipated, aligning with findings from the single injection timing sweep. The discrepancy in ID calculated from pressure ($\text{ID}_p$) and OH* chemiluminescence ($\text{ID}_{\text{OH}}$) between pure propane and the 90-10 blend is consistently less than 0.1 ms across all pilot timings. In contrast, the difference in $\text{ID}_p$ between propane and the 80-20 blend exceeds 0.4 ms for all pilot timings. This observation indicates a nonlinear change in reactivity with increasing DME blend percentage, highlighting that below a 10% DME concentration, the increase in reactivity is minimal. Figure 6.3 illustrates the ID results calculated from pressure and from OH* chemiluminescence as a function of pilot SOI.

It is important to note that for the two most advanced pilot timing (-50 CAD and -42.5 CAD), only the 70-30 pilot ignited before the main SOI. At the -35 CAD pilot timing, the pilots of both the 70-30 and 80-20 blends ignited before SOI, and for the two most delayed pilot timings (-27.5 CAD and -20 CAD), the pilot ignited before the main for all fuels.

Before continuing to explain the ID results, it is important to note that the method for determining $\text{ID}_p$ [47] was originally developed for single-injection CDC cases and is very sensitive to small increases in pressure due to low-temperature heat release. For the propane and 90-10 blend, minimal pilot heat release occurs before the main injection for the two most delayed pilot timings, although pressure rise and OH* chemiluminescence from the pilot heat release are detectable. This trend holds for the 80-20 and 70-30 blends, but the pilot heat release is more substantial, particularly evident in Figures 6.6 and 6.10. Consequently, the ignition delay (ID) calculated from OH* ($\text{ID}_{\text{OH}}$) is less than $\text{ID}_p$, consistent with the single injection intake temperature sweep trend. However, as the pilot timing advances, this trend changes, as illustrated in Figure 6.4.

At -35 CAD pilot timing, the pilot combustion produces less heat than at the more delayed
Figure 6.3: ID from pressure (left) and OH* chemiluminescence images (right) as a function of pilot SOI for all fuels. The error bars correspond to the 95% confidence interval about the mean.

Figure 6.4: Difference in ID calculated from pressure and OH* as a function of pilot SOI.
pilot timings, and it is highly premixed with weak OH* chemiluminescence (potentially indicative of low-temperature combustion chemistry), leading to \( \text{ID}_P \approx \text{ID}_{\text{OH}} \) for all fuels except the 80-20 blend, where \( \text{ID}_P > \text{ID}_{\text{OH}} \), following the initial trend. Further advancing the pilot to -42.5 CAD, where it becomes even more premixed, the pilot heat release does not generate enough OH* chemiluminescence before the main injection ignites, although the pressure rise due to the pilot heat release is detectable, resulting in \( \text{ID}_P < \text{ID}_{\text{OH}} \) for all fuels. Figure 6.5 shows an example of this early pressure rise without OH* chemiluminescence for the 70-30 blend at a pilot timing of -42.5 CAD. This effect is more pronounced in the 70-30 and 80-20 blends, which exhibit a more significant and advanced pilot heat release, often not discernible in the OH* chemiluminescence images at -42.5 CAD pilot timing.

Lastly, at the most advanced pilot injection timing of -50 CAD, the propane and 90-10 blend revert to the regime where \( \text{ID}_P > \text{ID}_{\text{OH}} \) because the pilot heat release is not detectable in the pressure before the main injection ignites, which is rich enough to produce detectable OH*, preceding the pressure rise. Conversely, for the 80-20 and 70-30 blends, \( \text{ID}_P < \text{ID}_{\text{OH}} \) at -50 CAD pilot timing, as there is still some detectable pilot heat release in the pressure before the main injection ignites.

6.3.2 Effect of pilot injection timing on propane and the 70-30 blend

To demonstrate the impact of DME blend percentage on cylinder pressure and heat release, Figure 6.6 presents pressure and HRR data for both propane and the 70-30 blend across all pilot timings. The heightened reactivity of the 70-30 blend is evident in the spikes in pilot HRR, starting around -10 CAD, which are absent in the propane HRR plots. Peak pressures were slightly higher for propane than for the 70-30 blend, primarily because of the more abrupt combustion when the fuel from the pilot and main injections burn simultaneously. At later pilot timings for the 70-30 blend, the pilot and main combustion events are clearly distinct. For example, with a -20 CAD pilot timing, the pilot heat release rate (HRR) peaks at over 40 J/CAD around -7 CAD, then drops to less than 10 J/CAD before the main
Figure 6.5: Raw pressure vs. CAD (top) and OH* chemiluminescence (bottom) from the same fired cycle using the 70-30 blend with a pilot timing of -42.5 CAD. The dashed lines on the pressure plot indicate the crank angle of the corresponding OH* images below.
injection ignites. This behavior contrasts with propane, where the pilot timing has a more pronounced effect on the main injection combustion timing due to propane’s lower reactivity and longer ignition delays.

In this study, injected fuel energy was kept relatively low to prevent high peak pressures and pressure rise rates. The short injection durations did not allow for a prolonged mixing-controlled combustion event typical in MCCI engines at higher loads. However, for the -20 CAD pilot timing, the SOC of the main injection preceded the EOI, which was determined from schlieren images to occur at -1.25 CAD. Therefore, increasing the duration of the main injection and increasing load would cause the HRR profile to approach typical MCCI with a longer mixing-controlled tail.

Due to the greater extent of premixing and lower reactivity of propane, the OH* chemiluminescence from the pilot combustion event is weaker than that of the 70-30 blend. This difference is evident when comparing the first row of images in Figure 6.7 and Figure 6.8, which displays single-cycle OH* chemiluminescence images from the -20 CAD pilot timing for propane and the 70-30 blend. It’s important to note that the timing of the images is not consistent between fuels due to the difference in combustion phasing.

In the images in row 1 of Figure 6.7, the OH* chemiluminescence from the propane pilot is barely noticeable at the periphery of the bottom part of the piston-bowl (large dashed circle) at -4 CAD. Moving to -2 CAD, the weak OH* chemiluminescence signal from the pilot is still visible around the periphery of the bowl, while one of the fuel jets from the main injection has auto-ignited before EOI. At TDC, fuel from all seven fuel jets has ignited, and the axisymmetric structure of the fuel jets is apparent. After TDC, the OH* chemiluminescence signal weakens due to the rapid depletion of available fuel as the expansion stroke begins.

The OH* chemiluminescence images of the 70-30 blend in the first row of Figure 6.8 exhibit distinct differences compared to the OH* chemiluminescence from propane combustion during the pilot combustion event. The 70-30 pilot ignites 5 CAD earlier than the propane pilot, accompanied by stronger OH* chemiluminescence. This advanced pilot ignition, com-
Figure 6.6: Pilot timing sweep results for propane and the 70-30 blend: (a) Cycle-averaged pressure and (b) heat release rate. Shading corresponds to the 95% confidence interval about the mean.
Figure 6.7: Single-cycle OH\(^*\) chemiluminescence images from the propane pilot timing sweep. Each row is labeled by pilot SOI timing, and the respective crank angle is in the upper left-hand corner of each frame.
Figure 6.8: Single-cycle OH* chemiluminescence images from the 70-30 blend pilot timing sweep. Each row is labeled by pilot SOI timing, and the respective crank angle is labeled in the upper left-hand corner of each frame.
bined with the higher reactivity of the 70-30 blend, promotes more advanced combustion of the main injection. Consequently, this reduces premixed heat release and facilitates further transition towards MCCI.

For propane, advancing the pilot start of injection (SOI) leads to a decrease in the maximum heat release rate and an increase in combustion duration. In contrast, the opposite trend is observed for the 70-30 blend, as shown in Figure 6.6(b). Additionally, Figures 6.7 and 6.8 indicate that the OH* chemiluminescence associated with the main injection combustion weakens significantly for propane as the pilot timing is advanced. In contrast, the OH* chemiluminescence for the 70-30 blend remains relatively constant, suggesting greater premixing for propane compared to the 70-30 blend due to its longer ignition delay. Furthermore, Rabl et al. [58] concluded that in combustion characterized mainly by diffusion, the burn period decreases with ignition delay, whereas in combustion characterized by mainly premixing, the burn period increases with ignition delay. This suggests that the main difference between the combustion of propane and the 70-30 blend under these conditions is that propane combustion is still mainly premixed at advanced pilot timings, while the 70-30 blend combustion is predominantly diffusion-controlled at all pilot timings.

6.3.3 Effect of DME concentration at constant pilot injection timing

The impact of DME blend percentage on the pilot-main injection strategy was assessed solely at the -20 CAD pilot timing due to the maximization of IMEP<sub>g</sub> for all fuels at this timing. For the 70-30 and 80-20 blends, the lowest amount of injected fuel energy was recorded at the -20 CAD pilot timing. In contrast, for propane and the 90-10 blend, the second-lowest amount of injected fuel energy was observed at the -20 CAD pilot, with the lowest occurring at pilot timing of -27.5 CAD. However, the relative increase in IMEP<sub>g</sub> from a pilot timing of -27.5 to -20 CAD exceeded the relative increase in injected fuel energy for propane and the 90-10. This implies that not only was IMEP<sub>g</sub> maximized, but so was the gross indicated
efficiency (GIE). Table 6.3 shows the IMEP\textsubscript{g}, injected fuel energy, and CA50, which refers to the crank angle at which 50% of the heat produced from combustion has been released.

Table 6.3: IMEP\textsubscript{g}, fuel energy, and CA50 at the -20 CAD pilot timing for all fuels.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>IMEP\textsubscript{g} [bar]</th>
<th>Fuel energy [J]</th>
<th>CA50 [CAD]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propane</td>
<td>2.56</td>
<td>528</td>
<td>1.00</td>
</tr>
<tr>
<td>90-10</td>
<td>2.50</td>
<td>510</td>
<td>0.50</td>
</tr>
<tr>
<td>80-20</td>
<td>2.33</td>
<td>494</td>
<td>-0.25</td>
</tr>
<tr>
<td>70-30</td>
<td>2.00</td>
<td>485</td>
<td>-0.50</td>
</tr>
</tbody>
</table>

The decline in IMEP\textsubscript{g} with increasing DME blend percentage cannot be solely attributed to reduced injected fuel energy. This is evident because the relative decrease in IMEP\textsubscript{g} exceeds the relative decrease in fuel energy as the DME blend percentage rises from 0% to 30%. Additionally, the combustion phasing, as indicated by the CA50, advances beyond TDC for the 80-20 and 70-30 blends. This advancement is known to lower thermal efficiency\textsuperscript{63}. It occurs because a significant portion of the combustion heat is released during the compression stroke rather than the desired expansion stroke.\textsuperscript{63} also showed that decreasing ignition delay in a Premixed Charge Compression Ignition (PCCI) engine led to lower efficiencies, which may also explain the decrease in IMEP\textsubscript{g} as DME blend percentage is increased.

Pressure traces for all fuels are illustrated in Figure 6.9. The traces for propane and the 90-10 blend exhibit striking similarity, with the 90-10 blend consistently advancing from propane by approximately 1 CAD. As the DME blend percentage is elevated to 20%, there is a noticeable advancement in combustion phasing, and a pressure rise occurs near -7 CAD that is not apparent in the 90-10 blend pressure trace. This trend persists as the DME concentration is further increased to 30%. Peak pressures decrease as the DME blend percentage rises from 10% to 20% and again from 20% to 30%.

The HRRs and cumulative heat releases depicted in Figure 6.10 indicate that the heat release from the combustion of the pilot becomes more significant with increasing blend percentage. This leads to a decreased HRR after the main injection because less unburned fuel is available to consume.
Figure 6.9: Fired-cycle-averaged pressure for propane and all blends at the -20 CAD pilot SOI timing. The shaded region corresponds to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.

Figure 6.10: Fired-cycle-averaged HRR (left) and cumulative heat release (right) for propane and all blends at the -20 CAD pilot SOI timing. The shaded regions correspond to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.
The cycle-averaged OH* chemiluminescence images before the main SOC and after the main SOC are shown in Figures 6.12 and 6.11, respectively. For the propane, the OH* chemiluminescence from the combustion of the pilot fuel is very weak and difficult to see but is noticeable at the bottom of the bowl at -4 CAD before the main ignites. The 90-10 pilot chemiluminescence is dim, but there is a noticeable signal at the bottom periphery of the bowl at -5 CAD before the main injection ignites. For the 80-20 blend, the pilot can be seen substantially igniting at -7 CAD, a significant advance from the 90-10 blend, and the 70-30 blend pilot begins chemiluminescing at -9 CAD. The ignition and combustion process of the main injection is similar for all fuels, with all 7 jets igniting and burning asymmetrically. There is a slight advancement of combustion phasing of the main injection as DME concentration increases, as expected with the increase in reactivity.

Figure 6.11: Cycle-averaged OH* chemiluminescence images of the combustion of the pilot fuel for all fuels at a pilot SOI timing of -20 CAD. Each frame is labeled with its corresponding crank-angle.
Figure 6.12: Cycle-averaged OH* chemiluminescence images of the combustion of the main fuel for all fuels at a pilot SOI timing of -20 CAD. Each frame is labeled with its corresponding crank-angle.
6.4 Conclusions

The pilot injection timing was varied from -50 CAD to -20 CAD in increments of 7.5 CAD, while keeping the main injection timing constant at -5 CAD. Intake thermodynamic conditions were maintained constant to isolate the impact of pilot injection timing. The results indicate a significant influence of pilot injection timing on combustion behavior, with the most efficient timing observed at -20 CAD for all fuels. In the case of pure propane, combustion remained predominantly premixed, with the pilot injection igniting just before the main injection. In contrast, for the 70-30 blend, the pilot injection released a substantial amount of heat before the main injection ignited, leading to a combustion process that was primarily diffusion- or mixing-controlled. However, the high (>50%) pilot mass fraction and the pilot igniting before TDC reduced the GIE for the 70-30 blend compared to propane. Therefore, optimizing the efficiency of the 70-30 blend may involve reducing the pilot fraction and shortening the dwell time. Advancing the pilot injection timing had a more pronounced effect on propane compared to the blends at advanced pilot timings, primarily due to the compounding effects of increased ignition delay and greater premixing.
Chapter 7

Glow-plug assisted ignition

7.1 Introduction

The ignition of propane and propane-DME blends containing less than 40% DME by mass in compression ignition engines is impeded by their low reactivity and prolonged ignition delays [4]. Although increasing the compression ratio can enhance combustion reliability, it may also result in elevated peak pressures and pressure rise rates, potentially damaging engine components. This necessitates a higher engine cost and significant weight penalty due to the need for stronger components to contain the increased cylinder pressures.

A potential solution involves employing a continuously operated glow plug (GP) to aid ignition. The use of GPs in compression ignition engines has a long history, dating back to the early 1920s [64]. While typically used to improve the cold-start performance of diesel engines [33][65], GPs have also been utilized in continuous operation to assist in igniting low-cetane fuels [29][66]. This chapter presents the results of the influence of a continuously operated GP on the ignition and combustion of propane and propane-DME blends of 10%, 20%, and 30% DME by mass. Continuing the methodology from the previous chapter, a dual-injection strategy was employed, with pilot injection timing varied from -35 to -20 CAD while keeping the main injection timing constant at -5 CAD. This timing sweep was
performed at two intake temperatures: 28 °C and 50 °C, representing light load conditions during cold and warmed-up states typical of a diesel engine.

7.2 Experimental details

7.2.1 Operating parameters

The intake and injection parameters for the 28 °C and 50 °C intake temperature conditions are displayed in Table 7.1.

Table 7.1: Operating parameters for the GP timing sweep that were constant for all fuels. The uncertainty for the isentropic temperature corresponds to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake temperature</td>
<td>C</td>
<td>28</td>
</tr>
<tr>
<td>Intake mass flow rate</td>
<td>g/s</td>
<td>4.52</td>
</tr>
<tr>
<td>Intake pressure</td>
<td>bar</td>
<td>1.06</td>
</tr>
<tr>
<td>Rail pressure</td>
<td>bar</td>
<td>1.12</td>
</tr>
<tr>
<td>Rail pressure</td>
<td>bar</td>
<td>500</td>
</tr>
<tr>
<td>Commanded injection duration (pilot/main)</td>
<td>ms</td>
<td>0.4/0.4</td>
</tr>
<tr>
<td>Pilot injection timing (SOI)</td>
<td>CAD</td>
<td>-35, -27.5, -20</td>
</tr>
<tr>
<td>Main injection timing (SOI)</td>
<td>CAD</td>
<td>-5</td>
</tr>
<tr>
<td>Peak motored pressure</td>
<td>bar</td>
<td>30.8</td>
</tr>
<tr>
<td>Peak motored isentropic temperature</td>
<td>K</td>
<td>838±16</td>
</tr>
</tbody>
</table>

This study investigated only the three most delayed pilot timings from the previous chapter. This decision was made due to the observed decrease in combustion efficiency associated with the two most advanced pilot timings. The same injection durations employed in the autoignition pilot timing sweep were also utilized in the GP study, ensuring consistency in mass across both experiments for the same timings. During the initial 10 firing cycles, only the pilot was injected, while in the subsequent 10 cycles, both the pilot and the main injection were injected. This separation allowed for the collection of pilot data uncontaminated by the effects of the main injection, providing a clearer understanding of their individual effects.
7.3 Results and discussion

7.3.1 Ignition delays

Figure 7.1 illustrates the ID of the pilot fuel jet interacting with GP, as determined from pressure and OH* chemiluminescence data under both 28°C and 50°C intake temperature conditions across varying pilot timings. Ignition would not occur at these intake temperatures with the GP turned off. When the fuel jet hits the hot surface of the GP, it triggers rapid ignition of the air-fuel mixture upon contact with the tip. This ignition event generates OH* in the proximity of the GP, which is detected by the OH* image processor. However, the heat release magnitude is low, and it takes several crank-angle degrees for sufficient heat release to occur for the pressure-based threshold to be crossed. Hence, a notable disparity exists in the ID calculated from pressure and OH* measurements.

Figure 7.1: ID calculated from pressure and OH* chemiluminescence vs. pilot SOI for the 28 °C (left) and 50 °C (right) intake temperature conditions. The shaded regions correspond to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.

The ID increases as the pilot SOI advances for both the pressure and OH* ID metrics, yet
the increase in ID derived from pressure is somewhat more pronounced compared to OH*. This discrepancy is attributed to the diminishing heat release rate of the pilot injection as the pilot SOI advances, as depicted in Figure 7.5. At advanced pilot timings, compressed gas temperatures are lower, reducing flame propagation and heat release from the pilot injection. Consequently, this yields a smaller pressure rise that is not as discernible in the pressure data.

The ID calculated from OH* is less affected by pilot timing, but there is a significant increase in ID for the propane and 80-20 at an intake temperature of 28 °C and for the 90-10 and 70-30 at an intake temperature of 50 °C. It was anticipated that blends with higher reactivity would yield notably shorter IDs. However, this expectation wasn’t universally upheld across all pilot timings, particularly evident in the ID calculated from OH*. The OH* ID metric is contingent upon a consistently low background signal, yet this background fluctuates from run to run, potentially influencing ID outcomes. Additionally, the OH* signal from the most advanced pilot consistently registered lower and exhibited greater variability for some cases compared to the delayed timings, introducing heightened uncertainty. Consequently, determining whether the observed increase in ID at advanced timings is genuine poses a challenge. Nevertheless, it’s crucial to emphasize that the average ID calculated from OH* at the -35 CAD pilot timing for all fuels remained within a range of 0.12 ms, representing less than 1 CAD at 1200 rpm. This underscores the independence of ID from fuel reactivity for the hot surface ignition process.

7.3.2 Intake temperature effect

Intake conditions wield considerable influence over ignition and combustion in CI engines due to their impact on ID, which is closely tied to temperature and pressure. Employing an ignition assistant such as a GP offers a means to mitigate the reliance on intake conditions for ignition and combustion while also moderating the influence of fuel reactivity by depositing energy into a fuel-rich zone to facilitate ignition. Despite the diminished reliance on intake temperature brought about by the use of a GP, comprehending the impact, or lack thereof,
remains vital. This section explores the impact of intake temperature on the GP-assisted ignition and combustion process of propane with a pilot SOI of -27.5 CAD.

Figure 7.2 illustrates the marginal variation in HRR between the two intake temperatures. Both the pilot-only and pilot-main runs exhibit minimal differences in HRR, with the only notable distinction occurring around TDC for the pilot-only case, where the confidence intervals do not overlap. On average, a higher intake temperature corresponds to increased heat release from both the pilot and main injection, as anticipated. However, the disparity in cumulative heat release depicted in Figure 7.3 is small.

Figure 7.2: HRR of the pilot-only cases (left) and the pilot-main cases (right) at both intake temperatures with propane as the fuel and a pilot SOI of -27.5 CAD. The error bands correspond to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.

**OH* chemiluminescence results do exhibit slight variations between the two intake temperatures, as depicted in Figure 7.4.** During the pilot-only cycles, there is a discernible uptick in OH* probability from -15 to -7 CAD at the bottom of the piston bowl near the GP. Contrasts between the two intake temperatures are evident throughout the pilot-main cycles, spanning from the start of the main injection SOC near -3 CAD to the conclusion of the image range at 9 CAD. Under the 50 °C intake temperature condition, 4 out of the 7
Figure 7.3: Cumulative heat release of the pilot-only cases (left) and the pilot-main cases (right) at both intake temperatures with propane as the fuel and a pilot SOI of -27.5 CAD. The error bands correspond to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.

Jets display non-zero OH* chemiluminescence probabilities at -3 CAD, whereas only 3 of the seven jets do so at 28 °C. This pattern of one additional jet structure exhibiting a non-zero OH* chemiluminescence probability at the 50 °C condition than at 28 °C persists across all pilot-main images, aligning with the HRR and cumulative heat release findings.

7.3.3 Effects of pilot SOI timing

Results from the pilot SOI timing sweep were evaluated using only propane as the fuel, allowing for the isolation of the effects of pilot injection timing on the combustion process. Figures 7.5 and 7.6 illustrate the pilot-only and pilot-main cycle-averaged HRR and cumulative heat release for all three pilot injection timings.

The -35 CAD timing exhibited the poorest performance, with the cumulative heat release of the pilot-only cases being less than half of that observed for the -27.5 CAD and -20 CAD pilot timings and just over half for the pilot-main cycles. The limited heat release observed at -35 CAD is attributed to the relatively low bulk gas temperatures at this early timing,
Figure 7.4: OH* chemiluminescence probability for pilot-only and pilot-main cycles with propane as the fuel, at intake temperatures of (a) 28 °C and (b) 50 °C a pilot SOI of -27.5 CAD. Image acquisition timing is constant for both intake temperatures and the corresponding crank angle is shown in the upper left-hand corner of each image. The concentric white dashed circles represent the injector and piston bowl and a small red dashed oval is centered on the GP near the bottom of the piston bowl.
hindering the ignition and combustion of the air-fuel mixture adjacent to the fuel jet in line with the GP. Furthermore, the fuel spray at this SOI timing is not all contained in the piston bowl around the GP as it is at the delayed timings due to the piston being further from the head and increased penetration rate. These effects are depicted in the top row of Figure 7.7(c), showing the small ignition zone of the pilot at -35 CAD compared to the pilot ignition zones of the -20 CAD and -27.5 CAD pilot SOI cases shown in Figures 7.7(a) and (b) respectively, which spread further around the periphery of the bowl. The substantial decrease in pilot heat release resulting from the latest pilot timing leads to inadequate combustion of the main injection, with high-temperature ignition only occurring in the half of the bowl nearest the GP.

The maximum HRR of the pilot-only increases as injection timing is delayed, owing to the higher bulk gas temperature at SOI. This leads to more rapid combustion of the fuel jet in line with the GP, as well as the air-fuel mixture from the adjacent jets. The cycle-averaged cumulative heat release for the -20 CAD and -27.5 CAD pilot timings is nearly
Figure 7.6: HRR of the pilot-only cases (left) and the pilot-main cases (right) for all pilot SOI timings at an intake temperature of 28 °C with propane as the fuel. Error bands correspond to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.

equal, with significant overlap in confidence intervals. Although the -20 CAD pilot timing slightly outperforms the -27.5 CAD timing in terms of pilot-only cumulative heat release, the pilot-main cumulative heat release for the -27.5 CAD timing is larger than that of the -20 CAD timing. It’s worth noting that the measured injected fuel mass for both timings is within 0.2 mg, with an uncertainty of 0.4 mg, indicating no significant difference in fuel energy.

The increase in pilot-main cumulative heat release from the -20 CAD timing to the -27.5 CAD timing can be attributed to the similar pilot-only cumulative heat release, coupled with lower in-cylinder density and a slightly longer dwell for the -27.5 CAD pilot. The lower in-cylinder density at SOI allows for faster penetration of the reacting jet near the GP, and after hitting the piston bowl, the reacting mixture and end gas travel farther toward the injector before the main injection due to the increased penetration rate and dwell. This is evident in Figure 7.7(a) and (b), where for the -27.5 CAD pilot-timing, there is probability of OH* chemiluminescence above the GP near the injector at -7 CAD, preceding the main injection,
Figure 7.7: OH* chemiluminescence probability for pilot-only (top rows) and pilot-main cycles (bottom rows) with propane as the fuel, at an intake temperature of 28 °C and pilot SOIs of (a) -20 CAD, (b) -27.5 CAD, and (c) -35 CAD.
while for the -20 CAD pilot-timing the pilot, the pilot-only OH* chemiluminescence occurs in a similar location at -1 CAD, notably after the main SOI.

7.3.4 Effect of DME concentration on GP-assisted combustion

The influence of DME blend percentage on the GP-assisted pilot-main strategy was evaluated at the -20 CAD pilot timing. Table 7.2 shows the IMEP\(_g\) as well as the coefficient of variance for the IMEP\(_g\) (COV(IMEP\(_g\))), defined as the ratio of the standard deviation of the IMEP\(_g\) to the mean IMEP\(_g\), expressed as a percentage.

Table 7.2: IMEP\(_g\) and COV(IMEP\(_g\)) for all fuels at the -20 CAD pilot SOI timing.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>IMEP(_g) [bar]</th>
<th>COV(IMEP(_g)) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propane</td>
<td>2.03</td>
<td>16.4</td>
</tr>
<tr>
<td>90-10</td>
<td>2.15</td>
<td>21.8</td>
</tr>
<tr>
<td>80-20</td>
<td>1.93</td>
<td>18.4</td>
</tr>
<tr>
<td>70-30</td>
<td>1.63</td>
<td>12.6</td>
</tr>
</tbody>
</table>

Other than the slight increase in IMEP\(_g\) from propane to 90-10, IMEP\(_g\) decreased with higher DME blend percentages. This decline can be attributed to the reduction in LHV as the concentration of DME increases. COV(IMEP\(_g\)) was well above 5% for all fuels, indicating high variability in the combustion event at this low-load condition.

Figure 7.8 illustrates the cycle-averaged results for heat release rate (HRR) and cumulative heat release. The cumulative heat release mirrors the trend of IMEP\(_g\) due to the similarity in combustion phasing among fuels. However, the substantial cycle-to-cycle variability complicates interpretation, as cycle-averaging diminishes much of this variability. Even with confidence intervals, deciphering the cycle-to-cycle variability remains challenging, and it is necessary to look at single-cycle results as well. The HRR from the cycles with the highest and lowest cumulative heat release for all fuels are displayed in Figure 7.9.

The pilot portion of the HRR for the highest cumulative heat release cycles exhibits a consistent pattern across all fuel types, characterized by a prominent peak occurring approximately at -12 CAD, with a magnitude of about 8 J/CAD and then decreasing rapidly until
Figure 7.8: Pilot-main cycle-averaged HRR (left) and cumulative heat release (right) for all fuels at a pilot SOI timing of -27.5 CAD with an intake temperature of 28 °C. Error bands correspond to the 95% confidence interval about the mean due to systematic error and cycle-to-cycle variability.

Figure 7.9: Pilot-main single-cycle HRR comparison for all fuels at a pilot SOI of -20 CAD. The left plot displays the HRR of the cycles with the highest cumulative heat release and the right plot shows the HRR of the cycles with the lowest cumulative HRR.
the main ignites. Conversely, the HRR after the main injection displays significant variability among the different fuel blends. Notably, the initial surge in HRR following the main injection is most pronounced in the 70-30 and 80-20 blends compared to others. However, following this initial spike, the HRR for these blends drops below that of the 90-10 blend and propane for the majority of the remaining combustion duration. In contrast, the initial peak HRR following the main SOI is less prominent in the propane and 90-10 blend, but there is another distinct peak shortly afterward, which, although rapidly declining, maintains a higher HRR compared to the 70-30 and 80-20 blends beyond the 20 CAD mark.

For the cycles with the lowest cumulative heat release, the pilot heat release shows more variation yet remains relatively consistent overall. The initial increase in HRR following the main injection mirrors the trend observed in cycles with the highest cumulative heat release, with the 70-30 and 80-20 blends displaying the highest HRRs, though this pattern is less pronounced. Subsequently, following the initial ignition of the main injection, HRR declines rapidly for all fuels until experiencing another brief surge from 5 CAD to 7 CAD before gradually tapering off as the expansion stroke progresses.

OH* chemiluminescence offers insight into the variances between high and low cumulative heat release cycles. Figures 7.10 and 7.11 display OH* chemiluminescence images corresponding to cycles with the highest and lowest cumulative heat release, respectively. In each row of both figures, the initial two images, captured at -2 CAD and 0 CAD, reveal similarities, depicting the jet aligned with the GP and the adjacent jets undergoing autoignition. However, for both propane and the 90-10 blend, small areas beyond the three jets closest to the GP exhibit OH* chemiluminescence at 0 CAD. By 3 CAD, significant contrast becomes evident: the high cumulative heat release images exhibit significant OH* emissions on the side of the bowl opposite the GP, whereas in the low cumulative heat release images, OH* remains confined to the GP side of the bowl. For propane and the 90-10 blend, ignition begins opposite the GP through autoignition. In contrast, for the 80-20 and 70-30 blends, ignition starts from the jets adjacent to the GP, and the OH* propagates to the opposite
side of the bowl. This phenomenon explains the absence of a spike in HRR after ignition occurs in the jets nearest to the GP for the 80-20 and 70-30 blends.

Figure 7.10: Highest cumulative heat release single-cycle OH* chemiluminescence images for all fuels at a pilot SOI of -20 CAD.

The ignition zones from the low cumulative heat release cycles fail to spread to the side of the bowl opposite the GP following the initial ignition event. Additionally, the OH* chemiluminescence notably diminishes after 6 CAD when flame propagation fails to occur on the opposite side of the bowl from the GP. This leads to the rapid decline in HRR after 7-9 CAD in Figure 7.9.
Figure 7.11: Lowest cumulative heat release single-cycle OH* chemiluminescence images for all fuels at a pilot SOI of -20 CAD.
7.3.5 Increased fueling

Based on prior research documented in [40], the inability of the combustion zone to propagate across the bowl from the ignition site was attributed to insufficient propagation of the end gas from the pilot towards the injector, thereby failing to ignite jets distant from the ignition point. This phenomenon is particularly noticeable under low load conditions when fuel energy is low. Higher load points were tested using propane and the 70-30 blend to validate this assumption. Both pilot and main injection durations were extended to 0.6 ms. In the higher load scenarios, injection timing was advanced to -27.5 CAD to allow sufficient time for the pilot combustion products to propagate toward the injector. To ensure fair comparisons, the low load results with the 0.4/0.4 ms pilot/main duration split presented hereafter also utilize the -27.5 CAD pilot timing. The increase in fuel mass and energy is detailed in Table 7.3.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Duration (pilot/main) [ms]</th>
<th>Mass [mg]</th>
<th>Energy [J]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Propane</td>
<td>0.4/0.4</td>
<td>11.2</td>
<td>520</td>
</tr>
<tr>
<td></td>
<td>0.6/0.6</td>
<td>17.3</td>
<td>800</td>
</tr>
<tr>
<td>70-30</td>
<td>0.4/0.4</td>
<td>11.9</td>
<td>490</td>
</tr>
<tr>
<td></td>
<td>0.6/0.6</td>
<td>18.1</td>
<td>745</td>
</tr>
</tbody>
</table>

Table 7.3: Injection masses and energies for the fueling study at a pilot SOI of -27.5 CAD.

The results depicted in Figure 7.12 display the HRR and IMEP\textsubscript{g} for both propane and the 70-30 blend at both fueling levels. Fuel energy was only increased by a factor of 1.5, but IMEP\textsubscript{g} increased by a factor of 2.3 for propane and 2.5 for 70-30. This substantial increase indicates a significant improvement in combustion efficiency.

Figures 7.13 and 7.14 depict the impact of increased fueling on the distribution of OH\textsuperscript{*} at various stages throughout the combustion process. Notably, there is a pronounced increase in OH\textsuperscript{*} concentration around the injector preceding the main injection and the elevated fueling. This indicates a greater presence of hot combustion products surrounding the injector at the main SOI, facilitating the autoignition of at least five of the jets for both propane and the 70-30 blend at the elevated fuel level. In contrast, at the lower fueling rate, only the three
Figure 7.12: Cycle-averaged (bold lines) and single-cycle (pale lines) HRR for propane and 70-30 at both fueling levels with the corresponding cycle-averaged $\text{IMEP}_g$.

jets nearest to the GP exhibit autoignition for all cycles. The relatively high reactivity of the 70-30 blend relative to pure propane is difficult to notice at both fueling levels, again indicating the lack of dependence of the GP-assisted combustion process on fuel reactivity.
Figure 7.13: Propane OH* probability maps at the low fueling level (top) and the increased fueling level (bottom).
Figure 7.14: 70-30 OH* probability maps at the low fueling level (top) and the increased fueling level (bottom).
7.4 Conclusions

This section examined the use of a continuously operated GP to assist the combustion process of propane and propane-DME blends at low-load conditions, representing both cold and warmed-up states of a typical diesel engine. Here are the major findings from the study:

- **Ignition delays:** At a GP power of 75 W ID is not strongly influenced by pilot SOI, intake temperature, or fuel composition. While both pressure and OH* chemiluminescence measurements showed an increase in ID with advanced pilot SOI, with pressure-derived ID showing a more pronounced effect, the changes are small relative to the ID. OH* ID measurements exhibit less sensitivity to pilot timing variations, although with some uncertainty due to background signal fluctuations. Notably, the average ID remains consistent across different fuels, highlighting its independence from fuel reactivity.

- **Intake temperature effect:** Intake temperature impacts GP-assisted combustion marginally, with higher temperatures generally resulting in increased HRR from both pilot and main injections. However, the cumulative heat release displayed only a slight increase with increased intake temperature. OH* chemiluminescence data indicated that with the higher intake temperature, there was an increase in the probability of ignition on the side of the piston bowl opposite the GP, which explains the increase in cumulative heat release.

- **Effects of pilot SOI timing:** Pilot SOI timing significantly affects combustion performance. A -35 CAD pilot timing leads to inadequate combustion of the pilot due to low bulk gas temperatures, hindering the autoignition of the jets adjacent to the GP. The combustion efficiency notably improved with the -27.5 CAD pilot timing compared to the -35 CAD timing. However, delaying the pilot to -20 CAD did not yield a similar enhancement in combustion efficiency. This lack of improvement can be attributed to
insufficient time for the end gas from the pilot combustion to propagate toward the injector before the main injection.

- **Effects of DME concentration:** DME concentration did not significantly affect the GP-assisted combustion as the hot-surface ignition is not heavily dependent on fuel reactivity. Although, the IMEP$_g$ and cumulative heat release varied with fuel. Higher DME blends generally lead to reduced IMEP$_g$ and heat release, attributed to the LHV of DME compared to propane.

- **Increased fueling:** Higher fueling levels enhance combustion efficiency, as evidenced by increased IMEP$_g$ and heat release rates. Elevated fueling levels result in a significant pilot heat release, leading to hotter bulk gas temperatures and more combustion products surrounding the injector before the main injection. This leads to a higher probability of autoignition of the fuel jets from the main injection and an increase in combustion efficiency.

Overall, these findings show that utilizing a continuously operated GP is a potential solution to facilitate MCCI of propane and propane-DME blends at low engine speeds and load. The timing of pilot injection significantly influences the performance of GP-assisted combustion, with the propagation of end gas from the pilot combustion crucial for enhancing combustion efficiency.
Chapter 8

Conclusions and future work

8.1 Conclusions

Propane has long been used as a fuel in SI engines due to its high octane number, but SI engines are generally less efficient than CI engines, and the potential GHG reduction is limited. DME is a potential candidate for use as a low-carbon alternative to diesel fuel in CI engines because of its high cetane number and clean-burning characteristics. However, due to its inferior LHV, limited supply, and corrosiveness towards common elastomers, the use of DME in engines also presents challenges. Blending propane and DME can overcome some of the challenges associated with pure DME, but even at DME mass fractions of 30%, the long ignition delays associated with propane’s low reactivity require high compression ratios and intake temperatures to reliably ignite propane and transition from high-pressure rise rate premixed combustion to MCCI.

This work utilized an optically accessible engine to investigate the fuel jet development, auto-ignition behavior, and the impact of a continuously operated glow plug on the combustion process of propane, DME, and propane-DME blends ranging from 10% to 30% DME by mass. Jet penetration and dispersion results did not indicate a significant difference in jet development between the different propane-DME blends, but there was a noticeable differ-
ence in dispersion angle between pure propane and pure DME associated with the difference in fuel density.

Near TDC, single-injection experiments demonstrated the influence of injection timing, intake temperature, and injected fuel energy on combustion. For propane and the propane-DME blends, ignition delays were on the order of 1 to 3 ms depending on injection timing, intake temperature, and blend composition, which prevented transition to MCCI. Utilizing a dual-injection strategy helped reduce high pressure rise rates and oscillations, transitioning the combustion to a mainly mixing-controlled regime. While the dual-injection strategy was beneficial in achieving MCCI combustion with propane and the propane-DME blends, high intake temperatures, and simulated compression ratios were still required for reliable combustion.

Using a continuously operated glow plug to assist ignition allowed for a significant reduction in intake temperature. Even at compressed gas temperatures representative of cold start CIDI engine conditions, the glow plug reliably ignited in-line and adjacent fuel jets. Increasing injected fuel energy significantly increased combustion efficiency due to greater pilot heat release and better propagation of the pilot combustion products towards the injector before the main injection.

Propane-DME fuel blends have the potential for use in CI engines. However, when DME concentrations are low, high compressed gas temperatures are needed for reliable combustion. This may require the use of intake heating, high compression ratio, or exhaust rebreathing which all would require significant engine modification. Another potential solution involves local energy deposition using a continuously operated glow plug, commonly used for cold start purposes in CIDI engines.
8.2 Suggested future work

Future optical engine studies of propane-DME blends should focus on investigating different piston bowl shapes and how they impact the GP-assisted combustion process. The use of custom piston bowls has been shown to improve the GP-assisted combustion process, which could extend the operating range and improve combustion efficiency. The GP-assisted studies in this work were conducted at relatively low in-cylinder densities, and future experiments should focus on assessing the performance of the GP at higher in-cylinder densities representative of typical CIDI engines. Another interesting GP study would be to use an injector with 8-10 holes instead of the 7-hole nozzle used in this work. The addition of holes would create a more uniform mixture around the periphery of the piston bowl which might help the pilot combustion propagate further.

The use of a pulsed laser ignition system should also be explored and compared to the GP. A laser ignition system was designed for the optical engine used in this work but has yet to be tested. As pressure increases, laser-induced breakdown of air requires less optical intensity, while forming an electrical arc using a spark plug requires a higher voltage. This makes laser ignition a potential solution to assisting the ignition of low-reactivity fuels in CI engines, but the effects of focal spot location, pulse rate, and the required laser energy to reliably achieve ignition are all unknowns that will require significant effort to understand.

This work focused on using propane-DME blends with 30% or less DME by mass. It would be useful to understand how the blends’ reactivity changes as the DME percentage increases over 30%. One potential approach would be to conduct single-injection autoignition experiments similar to those presented in Chapter 5, using 50-50, 40-60, and 20-80 propane-DME blends as well as diesel fuel. From these experiments, a propane-DME blend with a "diesel-like" reactivity could be determined. This would be useful because there is very little information on the CN of propane or propane-DME blends.
Appendix A

Delphi DFI3 operation information

Figure A.1 displays the necessary parameters for operating the Delphi DFI3 direct-acting piezo injector. The "IPhaseCurrentUpper" and "IPhaseCurrentLower" rows specify the maximum and minimum currents in Amps. At least 200 bar of fuel pressure must be applied before operating the injector to avoid damage to the piezo stack.

A nominal current level of 5 Amps is safe for operation at all pressures, but as injection
pressure is increased, the current level can also be increased. A current of 10 Amps is safe at a fuel pressure of 500 bar, and 12 Amps is safe at 2000 bar. The injection dynamics and rate of injection change with the current level so make sure to be consistent with settings. Figure A.2 displays a drive waveform for a 2 ms pulse width. The piezo stack is charged to 140 V when the injector is closed, so use caution when working near the injector while it is charged. If the maximum motored cylinder pressure is greater than 50 bar, ensure the injector is charged before closing the bypass valve to prevent the injector from opening unexpectedly.

![Figure A.2: Example drive waveform for a 2 ms pulsewidth.](image)
Bibliography


[28] Yao Liu et al. “Quantitative Measurement of OH* and CH* Chemiluminescence in Jet Diffusion Flames”. In: *ACS Omega* 5.26 (July 7, 2020). Publisher: American Chemical Society, pp. 15922–15930. DOI: 10.1021/acs.omega.0c01093 URL: [https://doi.org/10.1021/acs.omega.0c01093](https://doi.org/10.1021/acs.omega.0c01093) (visited on 05/08/2024).


